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THE ENGINEER'S PRACTICAL GUIDE,

and the WORKING of the

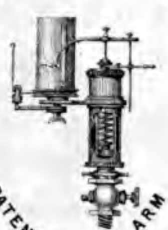
STEAM ENGINE

EXPLAINED

By the use of the

INDICATOR.

What the Stethescope is to the Physician the Indicator is to the skilful Engineer, revealing the Secret working of the inner System, and detecting minute derangements in parts obscurely situate.



PATENT SWIVEL ARM INDICATOR

BY

J. HOPKINSON & CO.,

HUDDERSFIELD.

Illustrated with Engravings & Diagrams.

AN EXPOSITION OF THE BEST MODE OF PRODUCING THE GREATEST AMOUNT OF POWER FROM A GIVEN QUANTITY OF STEAM WITH THE LEAST EXPENDITURE OF FUEL.

WITH A DESCRIPTION OF

THE MODE OF EXPANDING STEAM, AND THE COMPOUNDING OF ENGINES, WITH THE RATIONALE OF STEAM JACKETTING.

SEVENTH EDITION, ENLARGED AND IMPROVED.

THE Engineer's Practical Guide,

AND THE WORKING OF THE

STEAM ENGINE

EXPLAINED BY THE USE OF

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PUBLISHED BY
THE AUTHORS, BRITANNIA WORKS, HUDDERSFIELD.
1875.

PRINTED BY GEO. BRADLEY, HUDDERSFIELD.

PREFACE

THIS Work, although issued as a seventh edition, will be found on a perusal by those who have read the preceding edition, to be practically a new work, embracing only so much of the matter previously made use of, as is necessary and desirable for the treatment of the subject. The wide range of experience and observation which we have had in Steam Engineering, has enabled us, by investigation and comparison, to arrive at many important conclusions —some of which will now be made known for the first time, and which will be found fully and clearly set forth in the following pages.

The plan of this work is altogether different to that of any other on the Steam Engine. It is not intended to serve as a complete treatise on the Steam Engine, but to discuss those principles and features of the subject which are of the most practical importance for safe and economical working. The various chapters will sufficiently explain themselves.

The very large portion of the work which is devoted to the discussion of the subject of Compounding has been rendered necessary by the previously unsatisfactory and chaotic position of the question—the views advanced having been often vague and conflicting. Whether or not all the conclusions which we have arrived at may be confirmed by subsequent experience, we doubt not that the data and reasoning

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on the subject of Compounding will be acceptable to Engineers. It is the first time that this question has been discussed in any but the most fragmentary manner. The materials required for working out the solutions of the various problems of the Compound Engine have been derived from independent and original sources.

The chapter on Steam Jacketing is intimately connected with the one on Compounding, and may, in some measure, be regarded as a continuation of it.

The Appendix, containing the tabulated results of researches in many branches of physical science, will be found useful to Engineers and others interested in the subject matter exhibited.

At the end of the volume will be found illustrations and concise descriptions of a few of the useful Adjuncts of the Steam Engine and Boiler, which are manufactured by us. The object aimed at in the production of these, and of all other Boiler and Engine Mountings made by us, is to obtain the highest perfection of design, material, and workmanship, which the nature of the functions they have to serve can suggest.

J. HOPKINSON & Co.

BRITANNIA WORKS,
HUDDERSFIELD, JANUARY, 1875.

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THE WORKING ENGINEER'S PRACTICAL GUIDE.

CHAPTER I.

ON THE MANAGEMENT OF THE BOILER AND STEAM ENGINE.

THE Steam Engine has exercised the mental powers of almost unlimited number of persons for many years past, and great progress has been made in a knowledge of its principles. Much remains to be done to perfect it; and still more, to diffuse the knowledge already gained among those who are specially interested in its use.

More than 2000 years have passed since HERO, of Alexandria, for the first time in the recorded history of the world, obtained a continuous rotary motion by the agency of steam. Yet the growth the Steam Engine, as we have it now, is practically embraced within a century. Its importance to mankind it would be almost impossible to exaggerate, and scarcely could have been contemplated by the most prophetic of our forefathers.

The object of this work will be to treat the subject mainly from a practical point of view, and to diffuse such knowledge in a homely and understandable way as will enable the Practical Engineer, and those interested in the use of steam, to work it and produce the most economical results. We shall avoid all those abstruse disquisitions which only bewilder the brain, and really do not assist the practical man. Previous to the time of WATT the Steam Engine, although comprising the general principles of steam and vacuum, was of such a form and construction as to render its importance of no very great commercial value; but by the improvements of JAMES WATT, who may almost be termed its inventor, great strides were made, and to this day no material alteration has taken place since it left his hands. *There are thousands of* Engines at the present time con-

suming double the quantity of fuel they ought to, and this arises from a want of knowledge of the principles of the Engine, and the use of the Steam Engine Indicator, and as it is the province of this work to treat mainly on these topics, it is necessary to refer to the

MANAGEMENT OF THE STEAM ENGINE AND BOILER AND THE DUTIES OF THE ENGINEER.

The duties of an Engineer, entrusted with the management of a Steam Engine and Boiler, are of more importance than at first sight would appear. He ought to be attentive, and conversant with all the details of the constructions placed under his charge, as well as all the special circumstances of his particular case, otherwise he will be liable to make mistakes, and may unwittingly cause much damage.

Every part of the Engine should have his watchful eye, for the liability to derangement when working are so great that any direlection of his duty might cause costly breakdowns, as well as loss in working economy.

Boilers require careful and constant attention; but under the control of judicious managers, with proper fittings, they are perfectly safe, when properly constructed. But when their construction and management fall into the hands of incapacity and ignorance, or where reckless folly and hardihood have their full fling, death and destruction inevitably follow.

As this work is intended to be of practical service to the working Engineer, we shall, with familiar illustrations and examples, endeavour to make plain the several duties connected with his avocation.

THE GENERAL MANAGEMENT OF THE BOILER.

The Boiler and Furnace require the especial attention of the Engineer, these being the places where the power he has to apply and regulate is first generated, and from which the most serious consequences may arise, either from neglect or ignorance. The following short rules for good management ought to be deeply engraven on the memory of everyone entrusted with the care of Steam Engine Boilers, viz :—

I.—All dirt and lumber ought to be kept from the top of the boiler, and from every part of the boiler-house, and no person *admitted into the latter,* except on business.

II.—It should be ascertained regularly, that the Boiler is in good condition and properly stayed, where stays are necessary, particularly if fixed with angle iron; and the cotters (if any) kept up, and repaired as they are worn down; and it should also be seen that all the fixings attached to the Boiler are well riveted on, and good joints made with proper cement, free from injurious compounds detrimental to the metal of the Boiler. All leakages ought to be prevented, or stopped as soon as they are discovered, or they will soon destroy the Boiler. The flues and the surfaces exposed to the heat should be kept clean. Soot and dirt are non-conductors, and cause a loss of fuel, besides decreasing the heat-conducting power of the Boiler.

III.—See that the feed-valve is of good construction and kept in good working order, as from a defective feed-valve, serious results may occur. A leaky feed-valve is always a source of danger to the Boiler, and has often been the cause of serious accidents. In no case should it be placed below low water level; and see also that the float (if any) and glass-gauge are well attended to. The taps of the water gauge should be well tried at least twice a day, for if no low water safety valve is used, and this gauge is alone relied upon, the attendant may be easily deceived by the thoroughfares becoming choked, which is a very common occurrence, especially with some descriptions of water. If a self-regulating feeding apparatus is used, it should be closely watched, and on no account be implicitly trusted. Self-feeders to Boilers are liable to get out of order, from friction in stuffing-boxes, dirt, and other causes. Ascertain that the pipes and taps of the gauge are not made up with dirt or scale. If gauge taps alone are used, do not rely upon them to test the height of the water in the Boiler. If the water be below the tap, the moment the latter is opened to test the height, the water in the Boiler primes, and rushes to the tap, and shows water when the Boiler is actually short—particularly where great pressure is used. Gauge taps have been the cause of many Boiler explosions, from having deceived the attendant. Where a float is used, see that the float-wire is not strained, or worn to a shoulder, but kept free in the packing; and when worn smaller at the foot where the wire passes through the Boiler, see that it be renewed with a small wire—the smaller the better. Where steam is at a pressure above 10lbs to the square inch, the common float is of little use. The float-wheel appa-

ratus is an unreliable contrivance, and the sooner it is dispensed with the better. In most cases with high steam they are positively dangerous, and should not be trusted. Stop-valves, or other spindle-valves, should be constructed so that the thread on the spindle is *outside*, and may be seen at any time. This may be effected by means of a bridge-casting attached to the part through which the spindle-thread passes. Do not make the spindles and threads of any metal that will corrode and get fast, and always let them have such attention as they can be easily opened and closed in case of emergency.

IV.—Be provided with means to prevent the steam getting to a greater pressure than fixed upon, should the water from any cause become deficient, the safety valve should let off the steam to stop the working, independent of any control, until water is again supplied in the required quantity, thus saving the Boiler from injury or explosion. Often ascertain that the safety valve is properly adjusted, and in good working condition; avoid adopting all valves with pistons and complicated mechanical levers, which look well on paper, but are untrustworthy and unreliable in practice. Be particular that the spindles or guides (if any) have sufficient room to work, and also to prevent the parts adhering. Watchful care should be exercised when safety valves have large, flat, and beviled seatings, as they are liable to stick and become fast. Never adopt valves with outside flanges or mechanical means to cause them to lift higher from their seats: if such means have any power they retard their returning to their seats, at the proper pressure, and are thus a loss by blowing off unnecessary steam. In no case have a stuffing-box or spring to a safety valve. Safety-valves as commonly made and understood, are valves simply for letting off the steam. Thus constructed, they are in fact only steam valves, and not safety valves. A properly constructed safety valve will not only liberate the excess of steam when the pressure is too high, but also when the water in the Boiler from any cause becomes too low for safe working. A proper safety valve is also weighted in such a manner that when set at the pressure beyond which it is determined the steam shall not rise, the weight cannot be changed or tampered with whilst the Boiler is at work. Dead-weighted valves are superior to those having levers and balls, and are less liable to accident by interference, and not so easily tampered with by the ignorant and mischievous.

HOPKINSON'S Safety Valve answers all these requirements, of which a description will be hereafter given.

V.—Let the length of the safety valve lever be as short as possible, to prevent the extreme pressure you are working at from being increased with the same weight. When the lever of a commonly constructed safety valve is too long, the valve is easily tampered with. Indeed, the working pressure may, under such circumstances, be easily increased to, or beyond, the point of danger, by the weight being accidentally removed on the long lever. Tampering or removing weights on safety valve levers is very dangerous whilst pressure is in the Boiler, therefore should be avoided by all but practical and skilful men, and then only in cases where it is absolutely necessary. Let the joints of the fulcrum of the lever be of brass or some other metal that will not corrode, or cause the parts to oxidize and become fast. The fittings on the Boiler should be taken to pieces at least once a year, cleaned, refitted, and adjusted. This will prevent the parts becoming corroded and adhering to each other. Never rely upon lever safety valves placed on a pipe outside the boiler-house,—in frosty and cold weather they are positively dangerous,—if the valve leaks a little the escaping steam condenses on the parts and becomes frozen, thereby rendering the valve inoperative, and deceiving the attendant. Where large-sized lever valves (or valves of large dimensions) are used no reliance should be placed upon them—they are liable to stick, and invariably blow only from one side, and never discharge an equal quantity of steam to one of a proper size, and free to act.

VI.—Whenever from any cause the Boiler-plates become overheated, *open the damper and furnace doors to the full*, that the cold air may cool the Boiler. Closing the damper *increases* the heat in the flues, and by that means the danger is much greater. Lead plugs or fusible metal inserted in the flues, do not melt at the temperature required, after being in use a short time; they are positively untrustworthy; many Boiler explosions have occurred from the very causes these plugs were designed to prevent. The metal may become hardened, or undergo a change; they are liable to be covered with scale, or choked with soot (which are bad conductors of heat), therefore, place no reliance upon these contrivances; they only tend to lull the proprietor of the Steam Engine and the Engineer into fancied security. Above all, do not admit water into the Boiler c

relieve the pressure *suddenly*, as either will cause a violent agitation or foaming of the water—and this, washing over the heated plates, will generate steam and increase the pressure quickly—creating most imminent danger of an explosion. Under extreme circumstances of this nature, the water sent into the Boiler, or that brought into contact with the highly heated surface, will flash into high steam, tending to tear everything asunder. The starting of the Engine is quite sufficient to produce a violent agitation of the water, by the pressure being relieved when the Boiler is overcharged with heat, and disastrous consequences may be the result. Above all, do not depend on open column-pipes inserted in the Boiler, as a means of safety from explosions; for though they will relieve at times from over-pressure—to the great danger in such cases of running the Boiler empty—they are positively dangerous from another cause. It has been ascertained by experiment, that though the water in the Boiler may be reduced so low as to lay some of the plates bare to the action of the fire, the water from the pipe will not descend below a certain point, but will hang from the bottom of the pipe in the form of a cone—a striking illustration of what is known as the cohesion of adhesion. Should this cone, from any cause, agitation or otherwise, become detached from the end of the pipe, and be dispersed over the heated plates, the consequences above pointed out are almost inevitable. In fact, these open column pipes are no safeguard whatever against a Boiler becoming low in water, or, indeed practically empty. Where a proper safety valve has been applied, these open stand-pipes, fusible plugs, and also water-gauging taps are wholly unnecessary—as, with the former appliance (HOPKINSON'S valve) it is impossible for the Boiler to become short of water and have pressure within it at the same time; the prevention of danger from this cause being, under the circumstances spoken of, sure and certain.

VII.—See that the dirt in the Boiler is well blown out. Some descriptions of water will require the Boiler cleansing every three hours, others not so often, according to the quality of the water. If this point be attended to, the Boiler will wear much longer, and seldom require opening; the consumption of fuel will also be much less. Water impregnated with earthy matter forms a scum on the surface, which has a great tendency to cause the Engine to prime; in such cases periodical blowing off from the surface will be found of advantage: the deposit of scale in a Boiler is very injurious, and

many explosions have arisen from this cause alone. There is a great difference in the quality of water for Steam Boiler purposes, and too much attention cannot be paid to this point. Some waters scarcely give out any deposit, whilst with others the deposit is great; and this is a cause of much difficulty if not constantly removed. In Marine Boilers, the blowing out at stated intervals, from the surface and the bottom, prevents the deposition of salt and lime, and is the only thing required to keep the Boilers clean.

VIII.—Have the feed water for the Boiler as hot as possible, and sent into the Boiler *near the surface* of the water inside. The cold water will descend: but before arriving at the bottom it will have become heated, thus tending to keep the Boiler at an even temperature. If the water injected be low in temperature, it should be well distributed in the mass of water already in the Boiler, and not allowed to impinge on any portion of the metal of the Boiler,—or, that portion of the Boiler where the cold water thus comes into contact will be liable to crack and rend, from the continual action of contraction and expansion which such an arrangement inevitably causes. A pipe perforated with holes, carried for some distance into the Boiler, will be a sufficient means to accomplish this object. When the water is sent into the Boiler near the bottom, it does not rise until it has become heated; and this mode of feeding therefore tends to keep the Boiler bottom cooler than the other portions: an evil which ought to be avoided. The Cornish, Double Fire-box, and Tubular Boilers are more subject to this evil of unequal expansion and contraction than most others. The Engineer should therefore have the feed-pipes raised in the Boiler, when they are inserted in or near the bottom, otherwise the Boiler may break in the centre, and cause much loss or damage by the parts of the Boiler separating. With round Boilers, with the fire under the bottom, and particularly where they are hard fired, some mode should be devised to cause a continual circulation of the water near to the bottom of the Boiler; otherwise the steam generated at that place will partially displace the water, and thus allow the plates to become overheated and weakened. Especially is this the case with some descriptions of water, where there is a deposit of very fine dust, so that the Engineer should make himself acquainted with the particular description of the water, an its effects, and apply such remedy as suits the particular case und his care.

IX.—When there are two or more Boilers, with feed-pipes connected together, without a self-acting stop-valve between each, shut off the feed-valve to each Boiler during the night, or whenever it is not working: the water being liable otherwise to empty itself from one Boiler into another—as the pressure in the Boilers varies,—leaving one of the Boilers nearly empty; more particularly where there are Boilers working at different pressures. A self-acting stop-valve is in the last case absolutely necessary. But where HOPKINSON'S Patent Compound Safety Valve is on the Boiler, this will be prevented. Should the water lower, the valve will let off the steam, and the water will then be equalised, thus being what its name imports—a Safety Valve under all circumstances.

X.—Where junction steam-valves are used, or a valve placed between the Boilers and the main steam-pipe, see that the valve-spindle is one without a weight or screw, and take every precaution to prevent the valve being weighted down when the steam is up. Sometimes the safety-valve is on the steam-pipe; in that case junction valves are dangerous where they can be fastened down. In the stop valve for the Boiler, the spindle should be constructed with a screw thread and be outside. The nut for the screw can be fastened in the centre of a bridge cast to the valve-box lid. Where the screws are inside difficulties often arise. In all pipes fixed to the Boiler, have them so arranged that the condensed steam will drain back to the Boiler, and not go through the Engine; a great saving will be thereby produced—the hot water supplying the place of cold, according to the amount condensed. The arrangement will also, in many cases, prevent breaks-down, from the water of condensation getting into the cylinder.

XI.—Where there are two or more Boilers, do not keep any one out of action longer than is absolutely necessary for cleansing or repairs. A Boiler wears out faster when not in use, by oxidizing and corroding, than if moderately worked. It will be found more economical to work with "extra Boiler room," than to have one or more "standing." It will also tend to prevent "priming." The furnaces will be easier stoked by working a thick fire, allowing the [illegible] more time to give effect in the furnace. The furnace doors [illegible] be so often opened as with thin firing, the temperature [illegible] will be more regular, the cooling down whilst firing [illegible] contraction and expansion also correspondingly.

By this mode of firing, furnaces will last much longer, and in the case of a Fire-box Boiler, the plates in contact with the hot fuel will generate steam much quicker, with less labour and attention, there being an extra amount of absorbing surface in contact with the fuel, with the same area of grate,—and with some draughts a less area of grate is found to be better.

XII.—Where the Boilers are set in brickwork, *do not use lime in contact with the iron.* In setting the bricks use fire-clay, or common clay. Lime with damp quickly eats away the iron; therefore do not use it at all about the metal of a Boiler. Where a "midfeather" is placed under a Boiler, it ought to be of cast-iron, in the form of a **Λ**, for the Boiler to rest upon. Should any leakage take place, there will be little surface for the water to lodge on. Whenever the flues are cleaned, examine the exterior and interior of the Boiler thoroughly; and do this yourself. Do this also whenever the Boiler is let off. By keeping the surface of the Boiler and the flues clean, the draught will be improved, more steam will be generated in the same time, fuel will be saved, and the Boiler will wear longer. Be your own Inspector, and become what you ought to be—a Practical Engineer.

XIII.—Have a good steam-gauge to the Boiler: one that cannot be tampered with, and one not liable to get out of order. Do not place a tap between the Boiler and the gauge: for this, by being opened and closed, will interfere with the correctness of the indications. Fix the steam-gauge in a situation where the frost or cold cannot affect it. A column of mercury is the truest indicator, when used without a floating stick, or "steam peg." "Pegs" can be made longer or shorter to suit the convenience of the Fireman. There are various forms and makes of steam gauges; but many of them are nothing better than philosophical toys.

[In concluding this section, we would impress on all persons having charge of Steam Boilers, that there is no more certain sign of slovenliness than a dirty water-gauge or steam-gauge on the front of a Boiler, or the furnace-door covered with lime or dirt. Wherever these are met with, there is little hope of good management being found elsewhere under the same control.]

FEEDING STEAM BOILERS WITH WATER.

Too much consideration cannot be devoted to the importance of properly admitting "feed water" into the Boiler, and also to its being admitted at a proper temperature, for on these depend the durability of a Boiler, and the prevention of explosion from undue expansion and contraction, and certainly a great saving of fuel in the generation of steam.

In the first place, the person in charge should always ascertain that the water is at its proper height in the Boiler before the furnace fires are kindled, and that the water gauges (if in a fire-box Boiler) are placed at a proper level above the fire-boxes or flues at their highest point, so that none of the plates will be uncovered when water will run from the bottom tap of the water-gauge,—which has frequently been the case in Boilers where they are so constructed that the fire-boxes are in connection with an eliptical or other form of flue, where one part stands higher than the firing end, and frequently where Boilers are set at such inclines that the back or front of the fire-box may be bare when other parts are well submerged in the water. When Boilers are set to work filled with cold water, be careful to fire gently, so that the underside of Boiler may receive its proper proportion of heat, to prevent undue contraction and expansion of the top and bottom. Instances of rupture from strain thus brought about are very numerous; the writer has seen even new Boilers cracked across the bottom of the shell, where these instructions have been disregarded. The same evil arises in admitting cold water in vertical or other pipes, and delivering it at or near the bottom of the Boiler, for in such a case the tendency to unequal contraction and expansion is simply to produce a rupture at the part thus affected, and can result in no good in any sense. Cold water should at all times be avoided. The usual method was to take it from the hot well of the Engine, but it is now general to pass it through some description of water heater, known as an Economiser. With such an apparatus the supply of feed water to the Boiler is now admitted as high as 300° of temperature. There have been many contrivances for self-acting feed-valves, but none of them have proved efficient or safe to use. The simple method of having a hand or back-pressure valve capable of being regulated to any extent, and which lifts from its seat to admit the water, and closes at each stroke of the pump, to

prevent the water returning from the Boiler, is the best and most reliable means; and is generally fixed to the right or left side at the front of the Boiler, and the water conveyed some distance in by means of a pipe, perforated by numerous holes, so as to distribute the water in smaller quantities than would be the case if sent directly through, and delivering the contents at one particular place. The height or level of such pipes should be near the surface of the water, and not in any case or part thereof below that of the furnace crowns, for where no low water, or Compound Safety Valve, is attached, there is always danger of the valve being kept from its seat by dirt or other causes, then danger from overheating of the fire-box would ensue. Many persons apply a plug tap between the feed-valve and the Boiler, so that in the event of anything becoming wrong with the valve, the tap may be closed, and the valve can be taken to pieces or examined whilst the Boiler is still in operation; such tap thus permitting of that examination. Another great injury to a Steam Boiler is by admitting cold water to cool it for the purpose of cleansing, almost immediately it has ceased working.

The advantages of feeding a Boiler with hot in place of cold water is indeed great; but where a number of Boilers are used, say four or five, these advantages become more apparent; with that number the whole of the heat leaving so many furnaces, and impinging upon an economiser, concentrates the waste heat, proving so beneficial that one Boiler has been easily dispensed with. By all means see to the Boiler being covered with some non-conducting substance, and not left bare to the action of the atmosphere, and the radiation of the heat just generated. The practice of turning a brick arch and allowing the heat to pass over and around some few inches from the top of the Boiler, and from end to end, is becoming very general in some districts. Boilers are better when supported on brackets riveted to the sides and resting on the side walls, than upon midfeathers. In other districts there is a plan of suspending Boilers to iron arches on bridge castings. This system allows the contraction and expansion to take place without materially disturbing the brick work. Rollers also are employed to meet the requirement, to prevent the action on the brick work.

Various theories are entertained respecting the wisdom and safe of superheating steam when in contact with water, as is the case a Steam *Boiler over which* the products of heat may pass; but

this chapter precludes any solution, as attempted by many authorities, the reader will find the subject more fully treated under the head of Superheating and Steam Jacketing.

Betwixt the self-acting stop-valve and the pump have a tap, branching from the side of the pipe. When this tap is closed, the whole of the water will pass into the Boiler; but in case the water from the pump is more than required to supply the Boiler, the tap can be partly opened, so that one portion of the water forced by the pump may be sent through the small opening of the tap, and conveyed by a pipe to any required place, and the remainder forced through the valve into the Boiler. In proportion to the opening or closing of this tap, so will the supply of water to the Boiler be less or more. Another advantage of this arrangement is, that when no water for the Boiler is required, the tap can be set fully open, and the water will flow easily away, and offer no resistance to the action of the pump; or it can be forced into a cistern, to serve as a reserve for any purpose. The ordinary method is to have a weighted escape-valve, which requires power to force the water through—more power, in fact, than to feed the Boiler. The weighted valve also involves great wear and tear. A common inch tap will answer the purpose much better, at a twentieth part of the cost.

A small tap may be inserted in the pump bottom, or in any part of the barrel. When the pump is required to force its full quantity of water into the Boiler, the tap must be closed. If no water is required into the Boiler or cistern, open the tap. Air will then be admitted, and the pump will continue to work, without lifting or forcing water, and with very little power. This is all that is required to regulate the pump, and that too, without putting in or out of gear.

The "Injector," for feeding a Boiler by its own pressure of steam, is now largely used, particularly for Locomotive Boilers. The greatest drawback to the use of the Injector is, that the water used must be cold, or not more than 80° of temperature, otherwise the Injector will not act.

EXPANSION AND CONTRACTION OF BOILERS.

A great difficulty to be contended with in the management and working of Steam Boilers, arises from the unequal expansion and con-

traction of parts of the structure. In some instances these are so great as to be the cause of more "wear and tear" than any other process to which the Boiler is subjected. In the "getting up" of steam, therefore, great care should be taken, otherwise a Boiler may be seriously weakened in the process. An instance is well known where, from this cause, a Double Fire-box Boiler, when quite new, was broken right across the bottom, or under side, the first time the fire was applied to "get up" the steam, from the fire being urged too rapidly. This caused the internal flues to expand at a much quicker and greater rate than the outward shell; and the ends of the Boiler were thus forced outwards, until the strain tore the Boiler asunder. When the flues of Boilers of this form of construction are placed nearer to the bottom than to the top, the strain from unequal expansion and contraction is often such, that the plates of the under part of the outer shell are torn or broken; and in other cases leakages take place in positions where they are most difficult to discover.

In the "setting" of Boilers, all the surface possible should be exposed to the action of the heat of the fire—not only that the heat may be thus more completely absorbed, but that a more equal expansion and contraction of the structure may be obtained. With Round Boilers with "egg" or "dished" ends, and with the furnace underneath, too much of the surface of the Boiler cannot be exposed to the action of the fire and the heated gases. In some instances, where convenience serves, it will be found of advantage to pass the flue *over the top* of the Boiler, to equalise, in some measure, the heat, and consequently the expansion.

Where a Boiler of the last-named description is "set" with only a small portion of its bottom exposed to the heat, and a great portion of the structure exposed to the atmosphere, as is the common practice, a powerful action is thus left at full liberty to work out most injurious results. The heat will assuredly expand that portion of the Boiler to which it is applied; while the other portion exposed to the cold atmosphere will as assuredly contract. Thus the two forces are left to exert their respective powers *against* each other—tending to tear the Boiler asunder, by means almost imperceptible; and the ultimate result is not unfrequently a Boiler explosion, in spite of the fact that a structure in this form is the strongest possible, excepting the tru spherical.

In *the construction* and fixing of a Boiler, there are other mat

[illegible] require attention. Parts where the plates overlap should be [illegible] as little as possible to the direct action of the fire. Where [illegible] point is neglected, the outer plate is in danger of being burnt and [illegible], until in the end leakage is caused, and the plate is rent— [illegible] these tending to the destruction of the Boiler. Especially so is [illegible] case with the double-riveted Boiler, where intense heat is [illegible] to act upon the double-riveted portion. The high tempera- [illegible] causes the outer plate to expand more than the inner plate, and [illegible] firmly held between two lines of rivets, it cracks and blisters [illegible] thick or double-riveted portion, and this disturbs the rivets so [illegible] cause leakages; therefore double-riveting must only be carried [illegible] where the heat does not impinge at too high a temperature.

[illegible] years great improvements have been effected in the for- [illegible] and construction of fire-boxes and flues, by welding the lon- [illegible] seams, and turning up each edge of the plate, forming two [illegible] which are riveted together in the usual manner at their [illegible]. This plan adds immense strength to the resisting powers [illegible] against the collapse of the flue from over pressure, and [illegible] be formed more cylindrical, and thus approaching [illegible] greatest resistance, by dispensing with the lap joints. [illegible] which ought only to be placed in the hands of [illegible] Boiler Makers. We have seen some rude [illegible]; hence the remark as to whose hands [illegible] makers adopt a similar plan by welding [illegible] instead of turning a flange, a hollow ring [illegible] hollow railway line, is slipped upon each [illegible] thereto; thus having the vertical por- [illegible] only. Another plan is, forming a flue [illegible] the smaller of which is double flanged, [illegible] flanged on the edge, and the outer por- [illegible] with the body or line of flue; a plain [illegible] slipped on and riveted together, and [illegible] required length of flue is obtained. The [illegible] also stated by the makers to give [illegible] without bringing undue strain [illegible] expansion and contraction. It is [illegible]; and in like manner as in the [illegible] to the collapse of the tube from [illegible] Boiler apply vertical tubes in the

smaller diameter of flues for extra heating surface, and also for circulating the water, which is here particularly impressed upon all users should be a prominent feature in all Steam Boilers. For this purpose no better method has been devised than that of the conical tube—now so generally adopted for the Fire-box Boilers,—placed in different parts and positions of the flue, beyond the fire, and which also act in the double capacity of a circulator and stay. Immediately the heat begins to act upon the tubes, the water commences to circulate, and this materially assists in preserving the resisting power of the Boiler. Some makers now apply this form of tube, but dispense with the top and bottom flanges, and weld them directly to the flues.

The thickness of the plates of which a Boiler is composed is of importance in other respects besides that of requisite strength for the working pressure required. It is commonly supposed that a thin plate allows heat to pass through it into the water within, quicker than a thick plate can do. This is true, so far as getting the water heated to the boiling point. The Author has found, by direct experiment, that in a Boiler formed of plate one-sixteenth of an inch thick, water was brought to the boiling point in half the time it could be accomplished in a Boiler formed of plate three-eights of an inch thick, all other circumstances being the same. But he found, also, that when the fire in the respective cases had been three hours at work, the Boiler formed of the thick plate had evaporated more water than the one formed of the thin plate. Repeated experiments were invariably followed with the same results—showing the advantage of a moderately strong plate for the flues and furnace portions of a Boiler. The Author has found half-inch plate to be a desirable thickness for the fire-box and flues of Cornish Boilers, having regard to strength, economy and durability. It may not be amiss to observe also, that the quality of the iron, and the character of the workmanship, are also necessary things to be attended to in the construction of a Boiler.

Another matter to be observed in relation to a Boiler at work, is the unequal pressure to which its several parts are subject at the same time. Take, for instance, a Fire-box Boiler, working at 50lbs pressure to the square inch. That would be the pressure exerted on the upper side of the flues, slightly covered with water. Suppose the flues to be three feet in diameter, on the under side of those flues the pressure would be 51½lbs to the square inch—the weight or pressur

of the column of water having to be added to the steam pressure of 50lbs. It is from this fact, that the pressure is necessarily greater on the under side than on the upper side of internal flues, that, in cases of giving way from over-pressure, such flues crush *upwards* much more often than they crush *downwards*. There are also other actions going on in a Boiler of this construction, which add to this tendency —the strain and tension produced by unequal expansion and contraction. This is aided, too, sometimes by the ignorance of the Boiler Maker, of which an instance was given by one at Stockport, who constructed a Boiler with an oval internal flue, and who stayed the upper side of the flue, but not the under side, where the largest amount of pressure necessarily had to be sustained—probably from a notion that the only pressure to be provided against, at that part, was the weight of the water. The Boiler exploded—the flue crushing *upwards:* and seventeen lives paid the forfeit. However ridiculous it may appear to suppose that such a notion as the one described above could be ascertained by a Boiler Maker, the fact was sufficiently illustrated in the case of the Boiler explosion at the Castle Mills, Sheffield, some years ago. The first time that steam was "got up" in the Boiler in question, the bottom portion of the flat end at the back, below the flues, gave way, and the Boiler was sent like a rocket out of its seating, right across a public street, smash through some buildings on the opposite side of the street to the mills, and on into the river Don,—four lives being lost. On an examination of the Boiler, the Author found that the flat ends had not been stayed at the bottom portion, where the greatest pressure would necessarily be; and on the Boiler Maker being spoken to regarding this serious defect in construction, he excused himself by observing that he did not think the Boiler required staying in that part, because where there was no steam there was nothing but water, and consequently no pressure. Nothing possibly could be more absurd; and the two cases, out of many that could be adduced, show the necessity for all parts of a Boiler being made suitable for the various strains to which the structure is subjected when in use.

The accompanying engraving, Diagram No. 1, illustrates in a remarkable manner what has just been enforced—and illustrates also the tremendous effects produced on the structure known as the Boiler, by a Boiler explosion. The drawing represent the blown-out flue of a Steam Boiler, taken exactly as it lay after it had been sent with

artillery-force out of the Boiler, through the wall of the Boiler-house, and across a wide goit of water, into a passage on the other side, near to where there was another Boiler at work, which narrowly escaped destruction from the flying missiles—the large stone shown in the

Diagram No. 1.

TORN BOILER FLUE, AT UPPER ASPLEY, HUDDERSFIELD.

A. Interior of Flue—cylindrical when in Boiler. B. Form of under side of Flue, crushed upwards. C. Gusset Stay, attached to Flue.

foreground of the engraving having been sent through the wall of this second Boiler-house, with one of its corners impinging on the Boiler at work, producing a large and deep indentation. In this case *also, the flat ends of the Boiler* were only stayed on the upper

D

side. We shall subsequently refer more at large to this engraving, and the positions it illustrates.

This brings us to another subject connected with the working of Steam Boilers, viz.: whether the iron of which the Boiler is composed is stronger—that is, better able to resist tension or a crushing force when cold, or when heated up to the ordinary temperature of a Boiler when at work, according to the pressure of the steam—or from 212° to 300°, or more. At a Coroner's Inquest held at Leeds, on the occasion of a Boiler explosion in the neighbourhood, the following questions were put to an Engineer—a witness called to show the *cause* of the explosion: "Do you consider the Boiler to be stronger or weaker, when heated by the pressure and temperature of the steam?" Answer: "I consider that the Boiler is much weaker when the temperature is increased, than when cold." Question: "How much with steam at 50lbs pressure?" Answer: "Twenty-five per cent." It is lamentable that ignorance like this, though put forward with all the assurance of professional dogmatism, should pass current in a court charged with a most important and delicate inquiry—an inquiry into the cause of death consequent on a Boiler explosion. The very reverse of that deposed to is the fact; for iron, when heated to about 600°, increases in strength up to that temperature—that is, it increases in power to resist tensile strain, or to carry weight, up to the temperature named; and this is therefore held to be the point where the maximum strength of iron is attained. Thus the internal flue of a Boiler at work where the upper side is necessarily at a higher temperature than the under side, is the weakest in the latter part, as the result of two actions: first, less temperature—and second, higher pressure, as before shown. Here, then, in the matter of difference of strength arising from difference of temperature, we have another cause to account for the fact that Boiler-flues, in cases of explosion from over pressure, are more often crushed in from the under than the upper side. The two actions we speak of, which are continually in operation when a Boiler is in use, tend to force the flue out of the form in which it was first made—and thus, as it departs more and more from the perfect cylinder, to become weaker and weaker.

There have been many forms of apparatus for circulating the water in Steam Boilers, and different modes of setting, to produce [illegible], or rather to obviate the bad effects of unequal

expansion and contraction; indeed it is one of the first questions to be considered by anybody who may be constructing or putting down a Boiler.

The three diagrams here given, Nos. 2, 3, and 4, illustrate an improved Steam Boiler, patented by JNO. A. and JOSH. HOPKINSON, and was claimed to be quite a novel and curious method for accomplishing the object named. There is, no doubt, great merit in the invention, as, with little or no extra cost, it gives increased steam room, larger fire-boxes, anti-priming steam chamber, and an equal temperature top and bottom of Boiler. But in order that the reader may more fully understand it, we will give the explanation published at the time.

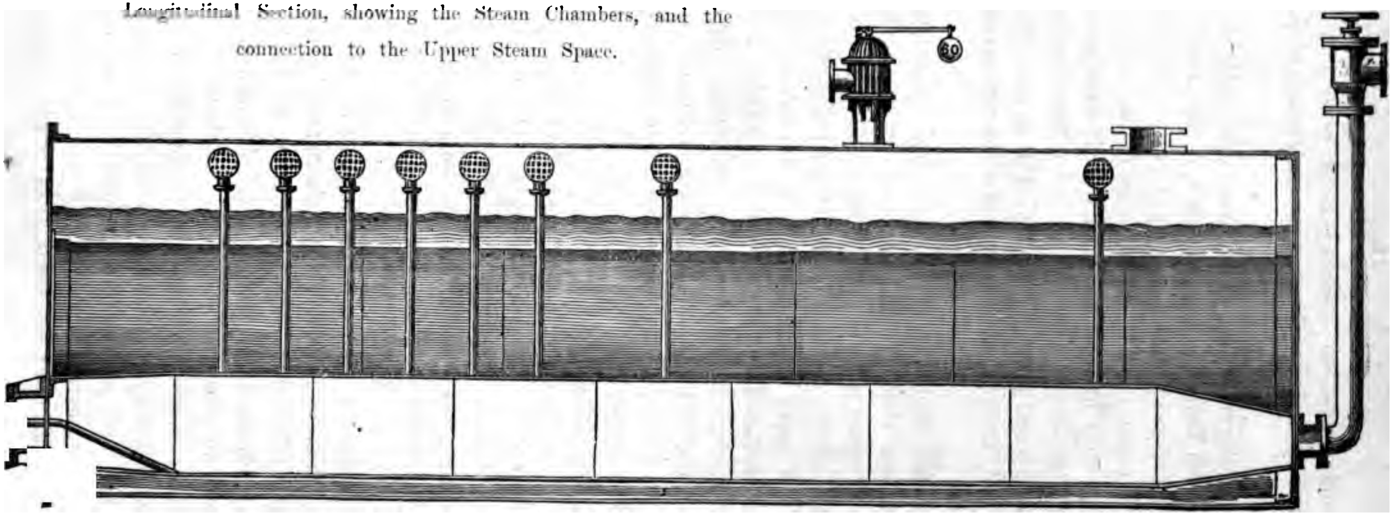
"The improvements in this Boiler consist in the application of a non-explosive steam chamber, placed under the fire-boxes, or in the water space, and extending from end to end of the Boiler. The steam is admitted into it by means of a series of pipes from that portion of the upper steam chamber to the steam chamber at the bottom of Boiler, and conveyed to the Engine either from the end or other part of lower chamber, as deemed desirable. By this arrangement the cutting of large holes, or the perforation of the shell for the admission of domes, is avoided,—which too frequently prove the cause of Boiler explosions.

"By this construction the fire-boxes are raised in the Boiler, which under the old method cannot be done without materially crippling the steam space above, and increasing the bulk of water below the fire-boxes. Water on the underside of flues is well known to be a source of destruction to Steam Boilers, from imperfect circulation; and regularly causes them to crack across the bottom, and produces leakages.

"The great body of useless water has also to be heated again and again—as the Boiler may cool from night to morning before starting, and at times of refilling,—which we venture to say forms no mean item of expenditure during twelve months' working. Now by this arrangement we substitute the steam chamber in the place of that useless water; or, in other words, we displace it, and therefore leave a much less quantity for the heat to act upon,—thus causing it to heat quicker, and promoting a circulation throughout the whole length of Boiler, thereby preventing unequal expansion by equalising as near as possible the whole contents of Boiler. Unequal contrac-

Diagram No. 2.

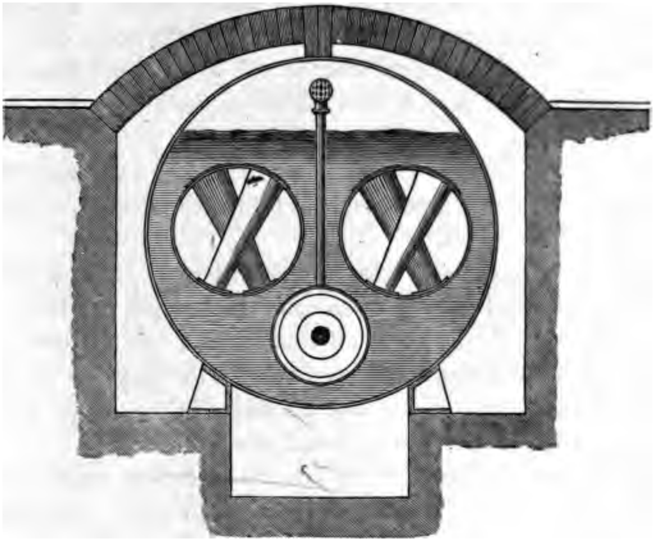
Longitudinal Section, showing the Steam Chambers, and the connection to the Upper Steam Space.



tion and expansion, we aver, is the principal cause of ruptures in Steam Boilers, producing leakages and corrosion, and entailing costly repairs.

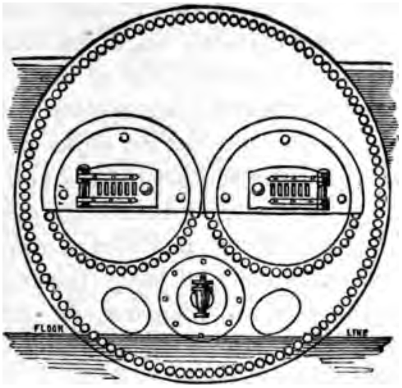
Diagram No. 3.

Section showing the position of Steam Chambers.



These Improvements are also applicable to ordinary Round Boilers without flues fired externally, and will be found advantageous by displacing the water and substituting steam.

Diagram No. 4.



Front Elevation.

"The fire-boxes, by being raised in the Boiler, will undoubtedly admit of their enlargement, and this we need only remark, is a feature in regard to economy of fuel which it would be advisable to put into

practice; it also greatly increases the power of a Boiler. This economy cannot be attained in the present Double Fire-box Boilers, without decreasing the upper steam chamber in capacity, and thereby increasing the probability of priming,—now found to be not only dangerous but a loss, robbing the Boiler of the highly-heated water, which should have been given out as motive power. Then again, by this arrangement we give increased strength to the ends of the Boiler, as it will be perceived that the new steam chamber also forms one strong tubular stay in like manner as the fire-boxes. By the addition of this tubular stay, or steam chamber, the whole three are divided more equally at the ends of the Boiler: each receiving its proportionate share of strain, and not being dependent, as is now the case, upon gusset stays.

"In many cases costly excavations have been required, so that the Boiler with the old form of dome could be brought to a lower level than that of the Engine, that all condensation might return to the Boiler instead of falling into the cylinder of the Engine; whereas it will be observed that several feet are gained in this respect, by constructing a Boiler upon the improved principle.

"In numerous instances we know that Boilers having on the ordinary domes, have been a source of great annoyance and expense, from the impossibility of passing archways on the high roads, and also from the inconvenience of fixing them upon their seats at the places where they have been intended to work.

"In the export trade the expense of transit has been enormous, from the fact of charging the whole length and breadth of Boiler, as measuring in capacity the number of feet on board vessel, simply in consequence of the projection of domes or other fittings.

"In this Boiler we have attained the object of applying a non-priming and anti-explosive steam chamber within the Boiler itself,—coupled with the fact of greater strength being given to the Boiler ends: the promotion of a better circulation of water in the Boiler: remedying the deleterious effects of unequal contraction and expansion: the enlargement of the fire-boxes, for the attainment of economy: and lastly, in the general improved construction for greater safety in working a Steam Boiler."

THE FORM, STRENGTH, AND STAYING OF BOILERS.

THE Double Fire-box, or Flue Boiler,—the form now in most extensive use,—is generally formed of three tubes: two lesser and one larger. The two lesser are placed within the larger, and all are bound together by end flat plates. The space around these tubes, and between them and the outer shell, constitutes the water space and the steam chamber. The pressure, whatever it is, which the steam exerts within the Boiler, is exerted *within* this space—that pressure being *internal* to the large or outer shell, and *external* to the smaller inner tubes or flues. That pressure, also, whatever it may be, is exerted upon the flat ends of the Boiler. This form of construction, which is the prevailing and favourite form, is not only not safe without careful staying, but positively and of necessity unsafe; and it follows as a matter of course, that with the high pressures now worked at, this want of safety becomes in many cases absolute danger; while with all Boilers of this construction, unstayed, and at whatever pressures they may be worked, if that pressure be appreciable, their giving way is but a question of time.

The great source of danger in this form of Boiler is the inner tube or flue, and the flat ends. When pressure is exerted within a tube or cylinder, with spherical ends, the tube can only give way by the metal being torn asunder; and the tendency of the strain is to cause the tube to assume the true cylindrical figure, or spherical form—the form of greatest resistance. With pressure exerted on the *outside* of a tube, the tendency of that pressure is to crush in the tube—to flatten it.

It is a well-known fact that iron of any strength, when formed into a tube, will bear a much greater strain to tear it asunder if that pressure be applied *internally,* than it will bear without crushing in when applied *externally.* A bar of iron when used as a tie-rod, will resist a very large amount of tearing force; but that same bar placed as a prop only under the weight exerted in the former case, would be doubled up and crushed out of form. The inner tube of a Boiler of this construction is but a series of props placed to sustain the immense weight of the pressure exerted externally to its diameter. The constant and never-ceasing tendency is for these props to give way—for the cylindrical tube to depart from the form of *greatest resistance,—to* become flattened or bulged; and its u

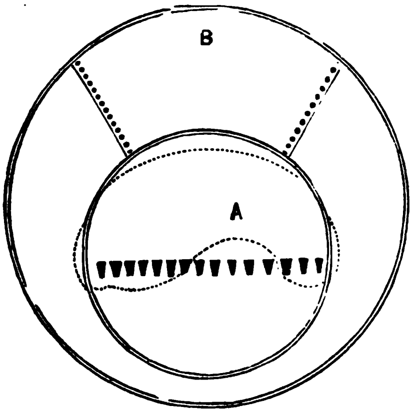
mate crushing in is, in the best of cases, where proper staying has not been resorted to, only a question of time.

The Double Fire box Boiler was invented in America by OLIVER EVANS, and was for a considerable period, and in some parts is still, known as OLIVER EVANS' Boiler. It was used in America so far back as 1786. Boiler explosions have been by no means uncommon in America, as is well known; indeed, so numerous had they been, and attended with such dire results, that in 1817 a searching inquiry into their cause, with a view to prevention, was instituted under the sanction of the American Government. In reference to the Fire-box Boiler, the report presented after that investigation states:—"Many respectable Mechanics and Engineers in this country considered that the improved Boiler invented by OLIVER EVANS obviated the objections to High-pressure Engines. The late melancholy occurrence on board the *Etna*, in the waters of New York Harbour, is evidence that they have been deceived." Mr. JACOB PERKINS, in his report on the same form of construction, says, "This form of Boiler should certainly be abandoned." Another eminent Engineer, Mr. C. J. JARVIS, wrote as follows:—"A flue of this kind may be placed in such circumstances, that when the steam and temperature get unusually high, it suffers a minute change of form. Under these circumstances it will inevitably collapse sooner or later, according to the extent to which its form is altered at each time it is unusually heated and the frequency of that occurrence, let it be surrounded with as much water as it may."

There is no doubt that, providing the flues were perfectly cylindrical, their strength would be very great; but such is not the case, nor is it possible to make them strictly true. The very weight of the material itself is sufficient to destroy the true figure of the flue, even providing it were made of one whole plate without rivets or lap joints; but in the ordinary flue, as used in all Boilers, the figure of greatest resistance is at once departed from by the overlapping of the plates. In the case of horizontal tubes, as employed in Steam Boilers, the pressure is not uniform; for, as has been already shown, while the pressure on the top part of a flue three feet diameter may be 50lbs per square inch, the pressure upon the under side would be 1½lbs more; because the weight of the column of water has to be added to the pressure on the lower part of the tube. What should be the *cylindrical figure in that* case is not the true figure of greatest resis-

tance. We therefore need not be surprised at the explosions which occur from collapse of flues, when not properly stayed. The very form almost seems to invite the occurrence. It is by a continued working of such description of Boiler, and the always increasing weakness of the flue, from its varied pressure and temperature, and its consequent change of form, that explosions ultimately occur. Hence the reason why a tube that has been tested to a pressure of 80lbs or 100lbs to the square inch, may afterwards fail under a pressure of less than half that amount.

Diagram No. 5.



CROSS SECTION OF BOILER, UPPER ASPLEY, HUDDERSFIELD.

A. Internal Flue, three feet diameter.
B. Outer Case of Boiler, five feet diameter.

☞ The dotted line on Flue shows the form the Flue assumed when crushed upwards, as shown in Diagram No. 1.

The accompanying Diagram, No. 5, is here given to illustrate what has just been advanced. It is from an exploded Boiler: the blown-out flue being shown by Diagram No. 1. The one now presented illustrates the general form of constructing the Cornish One-flued Boiler, showing the relative sizes of the two tubes of which the Boiler is composed, and also their relative position. A is the inner tube or flue, open at each end; B is the outer tube or case, closed at each end by the flat plates placed between the inner edge of B and the outer edge of A. The inner circle A shows the form of the flue

as originally made, and when at work. The small dotted line on the inner circle shows the form that flue assumed when it was ***crushed upwards*** by over-pressure, and sent out of the case like a tremendous bolt from a monster gun, torn, jagged, and rent in all directions, as shown by Diagram No 1. The two lines of larger dots, radiating from the outer edge of A to the inner edge of B, indicate the position of the two gusset stays placed at the fire-box end of the Boiler.

It has already been pointed out that there are other changes in connection with this form of Boiler, which cannot altogether be prevented from taking place. Plates crack and give way on the under side of the Boiler, from unequal contraction and expansion, caused by the difference of temperature of the top and bottom side of the Boiler. A Boiler also is materially hastened in its destruction by the emptying of it while hot, and then suddenly cooling it by admitting cold water, to get off the scale. Double Fire-box Boilers have been fractured on the under side, both in the line of rivets and even across the plates, from this cause. Observe also, when starting a Boiler and getting up the steam, and notice the length of time the water on the under side of the flue is before it is even lukewarm. Here, also, is an important action taking place in reference to the safety of the Boiler.

The flat ends of this form of Boiler are also a source of weakness. The reason of this will be at once apparent. The tendency of the force within the Boiler is to cause the flat end to bulge outwards—to assume, in fact, the spherical form. This brings unusual and unequal strain upon the rivets which join the plates together. These at last give way, being either torn out, or the plate itself is riven asunder across the line of rivets, and then out the ends go. Instances of this kind can be seen in almost every explosion. In some cases the plates have been torn asunder as though they were but paper. When once any part of a Boiler gives way, the other parts become exposed to unequal strain from the expanding contents, which exercise a tearing and impelling force equal to that of gunpowder.

To counteract this tendency of the flat ends bulging outwards, it is usual to *stay* them. Stays, at best, are but inferior substitutes for the form of greatest resistance. Stays would be of no service applied in a sphere subjected to internal pressure; the power of resistance would be exactly that of the metal to sustain the strain, exerted upon *all parts alike.* Stays could be of no advantage in such a construc-

tion unless they could be applied so as to strengthen that metal in all its parts; and this, as will be seen at once, could only be accomplished by using metal of greater thickness or strength for the original construction.

Boiler stays, therefore, are at all times but a substitute for real strength of construction. The manner in which they are frequently applied renders them insecure, and at times positively dangerous. The best plan of applying longitudinal stays is to let them proceed directly through each end of the Boiler, and then by means of washers, screws, and nuts to secure them in that place. The strain would then be direct to its own length, and its powers of resistance would be equal to the weight applied perpendicularly, which the iron rod would sustain without tearing asunder. This system of staying Boilers is now more generally adopted than was the case some years ago. The plan of longitudinal stays, applied as they were at that time, was positively unmechanical. They were bent at the ends to a right angle, and riveted at the outer portion to the Boiler end, thus giving the end freedom to bulge outwards by the force of the internal pressure, without securing any particular good beyond that of its own weight. There was also the rude way of fixing them to an angle iron riveted across the Boiler end, and securing the stay by a simple cotter. The continual action of the bulging wore the cotter to such a degree that the stays were altogether useless, and they remained in the Boiler as stays simply by name. As a practical test of placing longitudinal stays, and to prove how regardless the Boiler Makers were as to their proper application, the Authors fixed a dead-weight safety-valve upon a Boiler, leaving the suspended weight about two inches above the stay, but when the steam got to about two-thirds the pressure at which the valve should have discharged, the stays had become straightened out, and came in contact with the weight, thereby lifting the valve and discharging all the steam the Boiler could generate. It is an impossibility to fix them perfectly level, but the best way is the washer and nut; very frequently there are suspenders fixed to the Boiler top, so as to get them as rigid as possible. When we apply tie-rods to a beam of any kind, we should never think to copy the mode adopted with Steam Boilers; but should apply them so as to secure to the full their power of resistance against strain or weight; and so ought we to do in the case of Steam Boilers.

The principle of application appertaining to the gusset-stay is precisely that just described. It is formed, as before stated, of angle iron and a gusset of boiler plate. An accurate representation of the actual stays used in the case of the Aspley Boiler is given at C, in the engraving representing the crushed-in flue (Diagram No. 1). The strain in the case of the gusset-stays is on the angles of the angle-irons, and on the rivets or bolts by which these are attached to the ends and sides of the Boiler. The power of resistance is just the strain which these angles and rivets will bear without straightening, or breaking, or tearing out. It is not by any means the amount which the same metal differently applied would give. Samples of all the above recited modes of giving way were to be seen in connection with the gusset-stays of the Boiler at Aspley. In some cases the heads of the rivets were torn off, in others the metal was torn across the rivet holes, in others the angle was straightened, and in another the angle was broken, and appeared to have been so for some time.

Many years ago there were frequent cases of collapse of flues, arising from mal-construction. At the time when Mr. FAIRBAIRN launched before the public his patented improved Boiler, having oval flues, the late Mr. JOSEPH HOPKINSON not only lectured on the folly and danger of adopting such form of Boiler, but printed and circulated throughout Great Britain the following circular, warning Steam Boiler owners of the consequences that would ensue. To his energy and zeal in connection with the Fire-box Boiler, we may attribute the great improvements that have since been effected.

"The strength of a cylindrical tube to resist an external pressure exerted on its outward surface, is a very different thing from the strength of the same tube to resist the internal pressure.

"In the latter case, that is, when the force is exerted on the inside of the tube, and tending to burst or rend it asunder, the relative strength or power of resistance of the tube is very easily estimated; it is well known to be, under like circumstances, in the simple ratio of the thickness of the metal of which the tube is formed, and inversely as the diameter of the tube.

"But in the other case, when the pressure is external, the strength of the tube to resist such pressure will depend upon very different principles: it is generally supposed that the strength of a cylindrical tube under such circumstances must be immeasurably

great; and there is no doubt that such is really the fact, provided the pressure is uniform all round the tube, and that the true cylindrical figure is strictly preserved; because in such case the tube is like a well-formed arch; it cannot be destroyed except by the absolute crushing of the particles of metal one into the other, which is altogether improbable. But if the true cylindrical or circular figure is not preserved, and indeed if the deviation from the true figure of greatest resistance is ever so trifling, the principle of the arch is gone at once, and it is then like an arch without abutments; and the tube under such circumstance, instead of being able to resist almost infinite pressure, will in fact be unable to resist a comparatively moderate pressure.

"Now, practically speaking, it is almost impossible to form a tube that shall be strictly cylindrical or of any other figure of greatest resistance. The very weight of the material alone is sufficient of itself to destroy the true figure; the circumstances of the tubes of Steam Boilers being formed of metal plates with lap joints riveted together, precludes the possibility of obtaining the true figure; moreover, in the case of horizontal tubes, as they are employed in Steam Boilers, the pressure is not uniform; for while the pressure on the upper part of a tube six feet diameter may be only 13½lbs. upon the square inch, the pressure on the lower part of the tube will be nearly 16½lbs. to the square inch, because the weight of a column of water six feet high has to be added to the pressure on the lower part of the tube; therefore the cylindrical or circular form is not in that case the true figure of greatest resistance; and it is not very likely that in the ordinary way of business of Boiler Making, much care or correctness can or will be bestowed on. the calculating, or afterwards in the making of the tube, agreeable to the true figure of greatest resistance.

"Moreover, if all the above difficulties be overcome, and the tube is formed according to the true figure of greatest resistance, there is little chance that it will long remain so, in the practical working of a Boiler. The unequal contraction and expansion of the plates by being partially over-heated, and then suddenly cooled, accidents of constant occurence, will cause the plates to be drawn and buckled, and thereby soon destroy the true figure. And it should be borne i mind that however trifling the alteration of form may be at first, y when *once a slight alteration* has taken place, the destructive chan

then goes on in an accelerating ratio. And here a very important distinction should be observed, which is, that when the force is exerted within a tube tending to burst it outwards, the force exerted will not induce any change of form such as to render the tube weaker; because if the tube was originally made tolerably near to the true figure of resistance, any change of figure which afterwards takes place must be such as will render it in fact stronger—that is supposing the metal plates to have some degree of elasticity—it will cause the tube to assume the figure of greatest resistance. But it is not the same with a tube that supports an external pressure, because in this case any change of figure must demonstrably produce a greater departure from the figure of greatest resistance, and thereby render the tube weaker and weaker: and this is a very important reason why a tube that has been proved to a pressure of 80 to 100lbs. to the square inch, may afterwards fail under a pressure of less than half that amount.

"For the above reasons it is therefore clear, that a practical estimate of either the absolute or relative strength of tubes, supporting an external pressure, cannot be based upon the idea of these tubes being correctly formed, agreeably to the figure of greatest resistance; the safe mode to estimate the strength of the tubes by the capacity of the metal plates (of which the tubes are formed) to resist a transverse strain, in the same manner as we should estimate the strength of a flat plate, a bar, or a beam, the strength of which is known to be in the ratio of the square, of the thickness, or depth, in the direction of the strain.

"Under this view of the matter, therefore, it would be correct to consider that the strength of tubes under an external pressure would vary as the *square* of the thickness of the metal, but it is also clear that the strength will also vary *inversely* as the *square* of the diameter of the tube; because an increase of diameter not only increases the leverage, but also the absolute quantity of force in a like ratio.

"But it would not be right to suppose that the absolute strength of curved plates in a tube is no greater than that of perfectly flat plates; this is not the case. There can be no question that the curved form enables the plates to sustain a much greater force than flat plates are able to support; and it is clear that the nearer the curvature of the plates approximates to the figure of greatest resistance, the greater force they will be able to bear; for although, as is stated

above, the slightest deviation from the figure of the greatest resistance destroys the principle of the arch, and thereby reduces the comparative strength of such tubes from almost infinity, to a strength of very moderate limits; nevertheless curved plates will support a greater or less strain the nearer or more remotely they approach to the true figure of greatest resistance.

"It is therefore evident, that besides the capacity of resisting a transverse strain, there is also another element of strength in tubes subject to external pressure; that is, the strength derived from the curvature of the plates; but, as this latter strength depends entirely upon the greater or less approximation of the curvature to the figure of greatest resistance, and as the degree of approximation will vary in every individual case, and will also be liable to rapid alteration in the same tube, it is clear that no general rule can be given for determining the strength thereby gained. And, indeed, this is not requisite in the present instance, because it is not intended to offer a rule for estimating the positive strength of tubes, but simply the relative strength of different tubes, which, as above stated, is, in like circumstances, as the square of the thickness of metal, and inversely as the square of the diameter of the tube. The positive or absolute strength of such a tube can only be known by actual proof; but this once known, the strength of other tubes may be estimated by the foregoing rule:—thus, if a tube 3 feet diameter, and made of ½ inch plate is capable of sustaining a given external pressure, what will be the relative strength of a tube 6 feet diameter made of ¼ inch plate? Answer—The former is 16 times stronger than the latter.

"But, in case of the force acting inside the tube with a tendency to burst or rend it asunder, the strength of the tube will be as the thickness of the metal directly and inversely as the diameter; therefore if a tube is 3 feet diameter and made of ½ inch plate, it will be four times as strong as a tube 6 feet diameter, made of ¼ inch plate.

"The foregoing is a safe, easy, practical rule, and if employed by Boiler Makers in the planning of High-pressure Boilers, I believe it will be the means of preventing many serious errors and fatal accidents."

So far, this chapter has treated on the construction of Boilers, and the change of form produced by pressure. It may be well to show there are other actions the Fire-box Boiler is subject to when at work, *tending to destroy its* powers of resistance, and that however

r · …erly mounted to
r. … explosion. If a
… struction), and its
i … over-heated, it is
… sure it is capable of
… ing. It is therefore
l … working of Boilers,
… at the powers of resis-
… It must be understood
… -pressure,—whether that
… plates having become
… it is nevertheless over-
… now, may in two hours
… Boiler,—dealing death and
… water has painfully recorded
… this subject much attention
… s; and to the accomplish-
… only published thousands
… at great cost, through the
… called the attention of

… the principal papers of
… pages we deem is a fitting

… ABILITY TO EXPLOSION.

… the slightest knowledge of
… Boiler, that as it leaves the
… greater powers of resistance
… and that any departure
… ssarily reduce its resisting
… form or figure of greatest
… pressure, or, much worse,
… towards the giving way of
… ted to the varying pressures
… rated ratio until explosion
… vertible as the principle or
… other constructions where

pressure is external to the diameter (as is the case with fire-boxes and flues), it behoves each and all possessing such apparatus to prevent any such departure from the figure of greatest resistance.

"Many parties imagine that the flues of a Steam Boiler are similar in principle to that of an arch, but such an idea cannot be too soon dispelled, the very fact of the plates being riveted together with lap joints at once destroys the principle and takes from them the perfect cylindrical shape, which is the only true figure of greatest resistance. If, therefore, any Boiler of the fire-box construction be set to work imperfectly formed, or subject to greater pressure than the size and form justifies, though strong enough to begin with, it is but a question of how long the vicious strain can be endured, until at last, from the changing pressures and temperatures, it yields to the ordinary working pressure, releasing the highly heated contents, and resulting in serious explosion.

"Numerous instances of this form of Boiler could be shown, which were intended and would have worked for very many years with safety at the required pressure, had not a change of the form of flue been induced by the action of the furnaces, arising from a deficiency of water; whilst other Boilers have not suffered further than what is termed 'springing of the seams,' and by taking out the rivets, replacing them with new ones, and recaulking the plates, have impressed them with the idea that the Boiler was again perfectly safe. Such an idea cannot be too soon dispelled, and it is a duty we owe to those less acquainted with the subject, to announce the greatly reduced resisting powers of that Boiler, and, if space permitted, it would be easy of explanation.

"If, therefore, the evils arising from deficiency of water are fraught with such momentous consequences, what is to be done to render such occurrences impossible? We would refer the enquirer to the hundreds of parties who have applied 'HOPKINSON'S Patent Compound Safety Valve,' and having six to seven thousand in operation without either explosion or injury to Boilers; and if there is no answer to this fact, we plead it would be difficult indeed to convince.

"We, therefore, say to all Boiler Owners and parties putting down new Boilers, make yourselves secure by the adoption of this invaluable adjunct, both by night and day, and have the satisfaction to know that, so far as was in your power, you have done a duty to yourselves *and to those whom you* have entrusted with the care of

F

... mistake on the Engineer's ... spread on every hand."

... Boiler explosions may not only ... that they may occur from ... an explosion at the time, that ... and ultimately result in explosion ... of maintaining a Boiler in the best ... made manifest; and the following ... in the newspapers, will show that ... obtained at less cost.

... STEAM BOILERS COMPARED.

... has taken place, by which several lives ... twenty persons injured, besides a fearful ...

... by the authorities to examine and ... Explosion, state that 'the explosion ... of water in the Boiler.' Thus we ... Explosions have caused the ... injury to at least thirty ... worth of property.

... that to adopt our Patent ... of a Steam Boiler very ... guide, and calculate from the ... as recommended by some ... great efficiency and security ... less cost. For instance :—

	£	s.
... at 20/- per inch ..	9	0
... at 50/-	5	0
....	1	10
... taken out after use..	0	10
....	3	0
....	0	5
	£19	5

... on fittings, the whole of ... besides an annual outlay ... equal to five per cent. upon the

whole original cost; whereas to render a Boiler perfectly safe, either from over-pressure of steam or deficiency of water, the only fittings required are:—

One of HOPKINSON's Patent Compound Safety Valves,
One ditto Steam Pressure Gauges,
One ditto Water Gauges.

which we supply for £16 15s. Seeing, therefore, that safety can be combined with economy, it now devolves upon Boiler Owners to make choice of one of the two plans—the former attended with all the risks of explosion, and the latter rendering Boiler explosions impossible.

"We shall be glad to forward our descriptive pamphlet to any Boiler Owner.

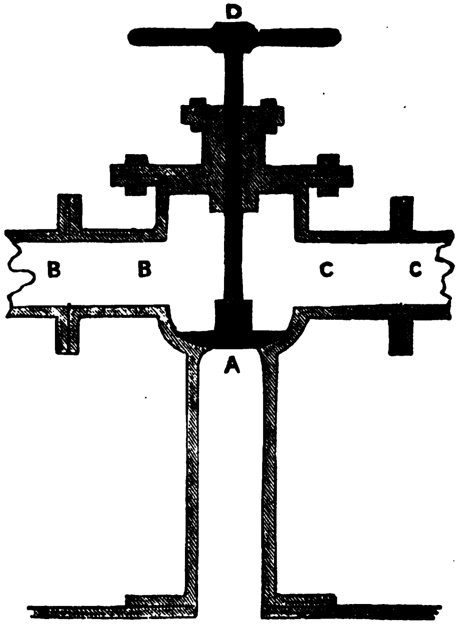
"Parties putting down new Boilers will, we trust, give due consideration to these facts, which, by being put into practical effect, will not only be the means of saving life and property, but keep from the hearths and homes of many families the misery occasioned by such lamentable catastrophes."

In a former page we warned the Engineer against the danger of having a stop-valve placed between the Boiler and the safety, or blow-off valve. Many Boiler explosions have arisen from this cause, of which the one at Upper Aspley, Huddersfield, involving a loss of twelve lives, was a memorable instance. On the top of the Boiler in question there was affixed a stop-valve, which is here represented in section by Diagram No. 6.

The lower lines of the diagram represent the Boiler top, with the pipe A standing upon it. To this pipe two arms, B B and C C, were attached, these also being pipes. B B led to the safety-valve, which was placed at a considerable distance from this point; and C C led to the Engine. At A, a stop-valve was placed, its seat being BELOW BOTH THE APERTURES, B B and C C. This valve was worked by a wheel and screw, represented at D. When the valve was open, the part which in our Diagram closes the passage at A, was raised up *above* the openings of the cross-pipes B B and C C, and then allowed the steam to pass either way—either to the Engine or to the safety-valve,—or to both. When the valve was closed, it was screwed down as now represented at A, completely "bottling-up the Boiler," and *preventing all escape of the steam*, except by rending the Boile

asunder? This was the exact state of this stop-valve—stop-valve up to a point indeed it proved to be!—and of the Boiler at the time of the explosion, and the state in which it had been for some time, perhaps for nearly an hour, before the explosion. This it was that prevented the steam from passing to the Engine, and induced the attendants to conclude that "the steam was down." Hence the injunction to "fire up," although danger then existed; and hence also the *firing*, until the limits of resistance within the Boiler had been passed. Hence also the catastrophe and its fatal results!

Diagram No. 6.



PLAN OF STOP-VALVE ON UPPER ASPLEY BOILER, IN SECTION.

A—Pipe proceeding from Boiler. C C—Feed-pipe to Engine.
B B—Pipe proceeding to Safety-valve. D—Wheel and Screw to raise and lower Stop-valve A.

The Inquest held on one of the twelve persons killed from the cause above detailed, was attended by Mr. WILLIAM FAIRBAIRN, C.E., in a professional capacity; and he on that occasion recanted, in a *great measure, his former* teachings in relation to the strength or

power of resistance of the internal flues of Boilers, with the pressure exerted externally to the diameter. His evidence on the occasion was: —"I may mention that I have recently been making some experiments with regard to the collapse of tubes. It was found that some flues collapsed unexpectedly, and when according to our ideas, they ought not to have collapsed. This induced a series of experiments interesting and conclusive. The Royal Society of London made me a grant for these researches. I took a Boiler of 36 feet, with a flue of three feet diameter, and another of 18 feet long, and I found that the long flue will go with half the pressure of the short one: that the strength is inversely as the length. I must confess that previously myself and other Engineers thought that the length made no difference at all; but I find that the strength of all Boiler flues, all tubes whatever, whether of iron or copper, follow a certain law, that the resistance is inversely to the length."

In relation to the explosion itself, Mr. FAIRBAIRN said in his report: "It will not be necessary to enter into calculations as to the forces generated in the Boiler, which led to such unfortunate results. Suffice it to observe that the flues and the ends were the weakest parts, and the first to give way. The former collapsed from compression on its exterior surface, and the fracture which gave vent to the elastic power of the pent-up steam was quite sufficient to account for the results that followed—namely, the demolition of the buildings, and the projection of the Boiler, flues, &c., into the positions in which they were found."

The result of a most searching, patient, and extended inquiry was the following verdict:—"The jury find that the death of Joseph Lum was caused by the explosion of a Steam Boiler, the explosion resulting from the screwing-down of a stop-valve, placed on the top of the Boiler, which when closed, completely cut off all connection with the safety-valve; but who closed that stop-valve, the jury have no evidence before them to show. But the jury, in returning this verdict, cannot but record their *strongest condemnation* of that combination of stop-valve and safety-valve which puts it in the power of any one at any time to prevent the action of the safety-valve by merely closing the stop-valve; and the jury consider that the Engineers who applied this dangerous construction are highly censurable, as is also the proprietor of the Boiler, who permitted it."

With *regard to the strength* of iron, the table which follows is

result of a series of tests conducted by Messrs. ROBERT NAPIER & SONS, of Glasgow.

STRENGTH OF WROUGHT-IRON BARS AND IRON PLATES.

BY MESSRS. ROBERT NAPIER AND SONS.

IRON BARS.

	Tenacity in lbs per sq. inch.		Tenacity in lbs per sq. inch
Yorkshire: strongest.. ..	62886	West of Scotland: weakest..	56655
,, weakest ..	60075	Sweden: strongest ..	48232
,, (forged)	66392	,, weakest	47855
Staffordshire: strongest..	62231	Russia: strongest ..	56805
,, weakest ..	56715	,, weakest	49564
West of Scotland: strongest	64795		

IRON PLATES.

Yorkshire: strongest lengthwise	56005	Yorkshire: strongest crosswise	50515
,, weakest lengthwise	52000	,, weakest crosswise	46221

NOTE.—The strongest lengthwise is the weakest crosswise, and vice versa.

According to these tests, Staffordshire bars were nearly equal to those from Yorkshire. Yorkshire Boiler Plates vary from the strongest, lengthwise, 56,005, to the weakest, crosswise, 46,221. These figures would represent the tenacity, or the resisting power of a Boiler, supposing that the plates were *welded* together; but as this is not practicable, one-half, or thereabouts, of the resisting power is *cut away* by the formation of rivet-holes. The strength of the Boiler is thus at once reduced to 23,110, or *half* the strength of the plate; and this, too, on the supposition that the rivet-holes in the plates are bored. In the case supposed, this rule would apply; but in the ordinary way of Boiler-making, the rivet-holes are not bored, but *punched* out of the solid cold metal by powerful machinery. This *punching* to a great extent *impairs the tenacity of the remaining portion of the plate betwixt the holes.* Then follows another process more injurious than the former—what is technically called "drifting." This "drifting" is a process whereby the holes of one plate are made to correspond with the holes of another plate, when the plates are in progress of being riveted together. When the holes do not exactly meet each other, so that the rivet will pass easily through both, the *Boiler-maker uses* an instrument called a "drift." This is a taper

"mandrill," which is driven into the two holes, and widens them out by the force of blows with the hammer. It often happens that the metal betwixt the hole at the edge of a plate is "cracked" with the "drifting"—in many cases weakening, and in others so far destroying the tenacity of the remaining metal, that the strength of the remaining portion cannot be taken at more than 20,000lbs. In some instances where the plate in its process of manufacturing has had but little cut from its edges, when being "squared up," the edges of the plate are quite brittle; and then, what with the before-mentioned mode of punching, and what with bending the plate in a cold state, such plate may be frequently seen "cracked" from hole to hole. From these and similar causes, a Boiler is often more than half destroyed before it leaves the Boiler-maker's yard.

In the process of bending Boiler Plates cold, the fibres of the metal are stretched, and their tenacity impaired or destroyed. Were they passed between bending-rollers, when heated to a proper working heat, the fibre would be properly drawn out, and its tenacity preserved—the same as in any other process of forging or shaping wrought iron. There cannot be a doubt that in the case of Boiler-flues bent cold to diameters varying from two feet to three feet, the plates are thereby weakened to a great extent. So long as the present mode of Boiler-making is pursued, we shall have weak and defective Boilers. The expense of heating the plates, for the purpose of bending them without injury, would be trifling. And there would be another resulting advantage: if there were any "flaws" in the metal, these would sooner show in the plate when in a heated state, than when bended in a cold state.

The suggestion has often been thrown out, that were the rivet-holes of Boiler plates bored instead of being punched, one part of the evil above pointed out would be remedied. There is no doubt but that process would be a great improvement upon the present practice; and if some independent firm would make Boilers with the same care and attention to material and workmanship as is done in Steam Engine construction, they would command a trade, although their prices would necessarily be higher than those who "make" in the ordinary manner. To the user, Boilers thus carefully built would be much cheaper in the end, to say nothing of the reduction to risk of life—the loss of which cannot be compensated for; and this to the humane is a *consideration of* importance.

The rule laid down for the strength of Steam Boilers, taking the tenacity of the plates at 60,000, is, according to the tests of the Messrs. NAPIER, far too high; and considering the before-mentioned methods of Boiler-making, we ought not to calculate the resisting power of Boilers at more than two-thirds the amount we have been in the habit of calculating. That is, instead of calculating the tenacity at 30,000lbs, we should not calculate that tenacity at more that 20,000lbs. Thus, the bursting pressure of a tube 30 inches diameter, made of $\frac{3}{8}$ inch plates, under the old rule would be 750lbs to the square inch. This new rule will be much nearer to the truth in practice than the former rule; and the tube would also require to be of good material, and well made, to resist 500lbs to the square inch.

Whatever may be the strength of Boiler plates before they are worked into shape, if care and attention are not paid to the bending, punching, drifting, and riveting, the best iron will become comparatively valueless. With the best material, the strength of a Boiler must greatly depend upon the manner in which it is made, and the care used in its construction, even if its form be correct in principle.

RULE FOR CALCULATING THE STRENGTH OF CYLINDERS OR BOILERS, AS GENERALLY APPLIED.*

THE tenacity of the metal of which a Boiler is constructed is about 60,000lbs, or six-sevenths that of good wrought-iron: a bar one inch square being the standard.

As, however, the cylinder which constitutes the Boiler is not whole, or *in one welded piece*, but is composed of a number of plates riveted together—the plates also being cut away for the holes—it will be necessary to diminish the number which expresses the tenacity. Let, therefore, the tenacity be put at 30,000lbs in place of 60,000lbs.

Multiply the numerator of the thickness by the tenacity of the metal, and multiply the denominator by half the diameter of the

* This rule is the one generally acted upon, and is no doubt pretty nearly correct, when the metal is without flaws, and the workmanship without defects. But as these cannot always be relied upon, the rule and the deduction from it will be found far too high for ordinarily made *Boilers.*

cylinder in inches; then divide the numerator by the denominator and the quotient will give the strength of the cylinder, or bursting pressure.

EXAMPLE I.

Take a cylinder 30 inches diameter, made of $\frac{3}{8}$ inch plate: thus—

$$\frac{3}{8} \times \frac{30,000}{15} = \frac{90,000}{120} \text{ then } 90,000 \div$$

$$120 = 750\text{lbs, the bursting pressure.}$$

EXAMPLE II.

A cylinder of six feet diameter, made of $\frac{1}{2}$ inch plate: thus—

$$6 \text{ feet} = 72 \text{ inches}$$

$$\frac{1}{2} \times \frac{30,000}{36} = \frac{30,000}{72} \text{ then } 30,000 \div$$

$$72 = 416 \text{ and 17-18ths lbs, working pressure.}$$

TESTING THE BOILERS.

The utility of testing Boilers to an extreme pressure, or to considerably more than that sustained in actual use, is more than questionable. Practice has proved that it is better not to test Boilers, beams, or bars of iron, with much more than the ordinary strain required. We may, by an extreme test, prove that the article has withstood such and such a strain; but we do not know how such a strain may have injured the substance tested. For instance, a cast-iron beam was tested at the foundry of Messrs. Millburn's, Staleybridge. The beam was 27 feet long. A weight of 24 tons was hung on the centre, with which it deflected $1\frac{1}{8}$ inches. When the weight was removed, the beam assumed its proper form. The same beam was afterwards tested with 16 tons hung on the centre, when it broke—though there were eight tons less than it had withstood before.

Another instance was afforded in the test of a Steam Boiler at Messrs. Hargreaves, at Accrington. The hydraulic test was resorted to. The extreme working pressure of the Boiler was a little over 40lbs to the square inch, and the test the Boiler was submitted to was 70lbs to the square inch. To all appearances, under the test the Boiler was safe at that pressure. In about three months after this *testing, the Boiler* exploded; and on investigation, there

was sufficient to show that the test to which the Boiler had been subjected had injured the structure—and from that time till the whole gave way, the Boiler had been gradually getting weaker. The flues had been strained by the test, the pressure being on the outside of the diameter.

There can be no doubt that the testing of any structure or machine to the full strength the apparatus may be expected to resist, is highly necessary ; but beyond *that*, there is more risk of injury to the apparatus tested, than any chance of good resulting from an extreme test.

The word " test" carries with it to the theorist and the casual observer an idea of security—because such test affords proof that the machine or structure has withstood a certain force. This may be quite true : but it does not follow that the same machine or structure will resist the same amount of force again, or anything approximating to it.

CLEANSING BOILERS.

It is well known that water impregnated with earthy matter produces, when boiling in a Steam Boiler, a scum on the surface. If this scum be collected and blown out, the Boiler will seldom require to be opened and cleansed ; but if the scum be allowed to accumulate and settle upon the plates where the fire impinges, the injury to the Boiler will be in proportion to the thickness of the accumulation, which, according to the nature of the deposit, assumes the form of scale or mud. When it assumes the first-named form, the Boiler will require to be often opened and cleansed, or the injury from the burning of the plates will be great.

There is a great difference in the quality of water for Steam Boiler purposes. Some waters scarcely give out any deposit, while with others the deposit is great; and this is the cause of considerable difficulty if not constantly removed. It therefore behoves the Engineer, when he has to work with water of the latter description, to be exceedingly attentive, and to look well after the interior as well as the exterior of his Boiler; and particularly where the water has spent dye-wares and acids mixed up with it, as is often the case in manufacturing districts. In this matter, as in many others, he may lay

it down as a rule, that the utmost cleanliness possible will result in a saving of fuel, and in the prevention of much "wear and tear" to the Boiler.

THE SAFETY VALVE.

THE form and construction of this indispensable adjunct to the Steam Boiler are of the highest importance, not only for the preservation of life and property, which would, in the absence of that means of "safety," be constantly jeapordised, but also to secure the durability of the Steam Boiler itself. And yet, from the manner in which many things called Safety Valves have been constructed of late years, it would appear that the true principle by which *safety* is sought to be secured by this most valuable adjunct, is either not well understood, or it is disregarded by many Engineers or Boiler-makers. Many of those unfortunate calamities—Boiler explosions—have occurred when, to all appearances, the Safety Valves attached have been in good working order: and Juries under the presidency of Coroners, have not unfrequently been puzzled, and sometimes guided to erroneous verdicts, by scientific evidence adduced before them, tending to show that nothing was wrong with the Safety Valves—and that the devastating catastrophes could not have resulted from over-pressure, because in such cases the Safety Valves would have prevented them.

If Dr. PAPIN,* the inventor of that most useful and scientific apparatus, the Safety Valve, could but witness some of the forms given to the instrument, and some of the modes of construction adopted, he would indeed marvel to see the degeneracy which, in this day of general improvement, is but too often apparent in the manufacture. and construction of this essential and indispensable adjunct.

The portions marked 1 and 2 on the accompanying diagram, page 45, represent variations of the Common, or Mushroom Valve. This is usually constructed with a guide-pin, to pass through a hole or socket provided in the cross-bar inserted in the seating of the valve—such guide-pin and steadying-socket being *underneath* the valve,

* Dr. Denys Papin, a native of Blois, in 1680, invented the common Lever Steam Valve, misnamed the "*Safety Valve.*"

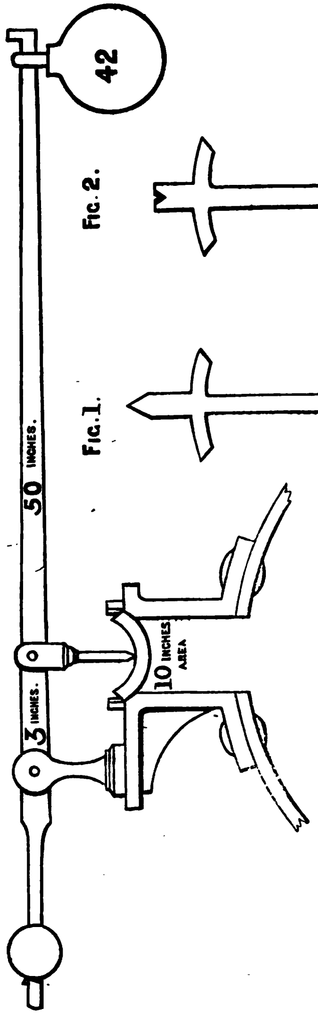
and consequently *within* the Boiler, constantly and fully exposed to the action of the steam, the varying temperature, and to the accumulations of dirt and other extraneous matters. With this form of construction alone, numerous Boiler explosions have occurred—from the stem, or guide-rod, of the valve having got bent, or otherwise damaged; from its having become corroded in the steadying-socket of the cross-bar; from its becoming fast in its place, when closed; or from many other and obvious causes. It is clear that when the Mushroom Valve is in any one of these states it is impossible for it to act, and the Boiler is then, to all intents and purposes, as if nothing in the shape of a Safety Valve had been provided.

Another objection to this form of Safety Valve is, that the weight to hold down the valve till the limit of safety is approached, is applied in the worst form possible to ensure accuracy in the indications of the valve, or to allow of the free working of that valve when over-pressure requires it to open. In valves of this construction, there is generally provided on the lever a swell, or projection, rounded down at the end to a dull point; and this dull point rests in a small counter-sunk hole in the outer tip, or central projection of the valve (as in the portion numbered 2 of the engraving, page 45); or the point on the tip of the valve rests in the counter-sunk hole in the projecting portion of the lever (as in the portion of engraving numbered 1). When the weight is hung on the other end of the lever, or at such place upon it as has been arranged for, the swell, or projection, on the lever bears upon the tip of the valve, and keeps that valve down on its seat until it is raised therefrom by over-pressure within the Boiler. But it is a question arising from this mode of applying the weight, whether the valve *can* act at the time and in the manner designed. In the first place, the joint at the end of the lever, where it is attached, must necessarily have some play given it. Oftentimes greatly too much is given; or rather, no care to have exactness of fitting is exercised. This play, from the action of the lever and wear and tear, becomes greater and greater; and every departure from exactness, either in the original manufacture, or from subsequent wear, takes the weight from the *centre* of the valve, and brings it more upon one or other of its sides than the rest. These deviations from exactness may be small, almost infinitesimal, in themselves: but multiplied as they are in fact, in the length of *the lever from its* attachment to the point of bearing on the valve,

they become very appreciable, and highly detrimental to the free action of the valve. When the valve is thus weighted out of its centre, two separate but injurious actions take place: first, when

Diagram No. 8.

LEVER SAFETY VALVE.



over-pressure occurs, instead of the valve being lifted bodily from its seat, it becomes tilted, and the steam escapes at *one side* only; and second, in consequence of this tilting, the guide-rod often becomes bound *fast in the guiding*-socket of the cross-bar, and the valve

thus prevented from operating, except in a very limited degree—not nearly sufficient in case of over-pressure. The worst is, that under the circumstances supposed, the higher the pressure—that is, the greater the danger from which the valve is required to relieve—the less capable does it become of operating in the manner for which it was calculated and designed: for the greater the force brought to bear upon the tilted portion of the valve—(and be it remembered that this tilting exposes more of the surface of the valve-face on the tilted side than on the side which still remains covered by the seating) —the more firmly does the valve become bound in the guiding-socket. From these and other causes of a similar nature, Safety Valves of this construction cannot be made to work accurately—and they are totally inapplicable for high pressures, if "safety" is to be a consideration. The indications, however carefully the valves may have been weighted, are often most inaccurate. The Authors have known valves of this description to be kept as perfectly down to their seatings as they were capable of being kept—that is, without any escape of steam from over-pressure—when the pressure in the Boiler, as measured by a more accurate test, has been 12lbs more than the valve was calculated and weighted to blow off at. With a valve of this description, anyone, by moving the lever to and fro, may cause the steam to escape at either of its sides. To counteract this tendency to tilt, other guides have at times been applied; but it has been found in practice, that this attempt to remedy one evil has only been to create others, which equally tended, and in like manner, to prevent the proper and safe working of the valve.

The main portion of the accompanying engraving (Diagram No. 8) is a representation of a Safety Valve which has been designed and introduced to prove that a valve of the ordinary kind may be made to act well, generally speaking—to blow off from over-pressure, with far greater accuracy than either of the constructions before considered, and also with a uniform discharge of steam from all sides of the valve orifice, when open. The valve is of the spherical form on its under side; and it is placed within outward guides affixed to the seating. The weight is brought upon the valve from a low-fixed centre—the centre-pin being jointed to the lever, or fixed thereon by other means, so that there be the requisite play for the pin always to bear on the Spherical Valve. The small weight on the short or *fastened end of the lever,* is merely for the purpose of balancing the

long end of the lever, so that when the ball-weight]for weighting the valve is off, the valve is entirely free from outward pressure. Where this portion of the arrangement is dispensed with, the weight of the lever as it bears on the valve has to be taken into account, when calculating the weight to be used for weighting the valve, and also the several distances on the lever at which that weight will have to be placed for the various pressures required.

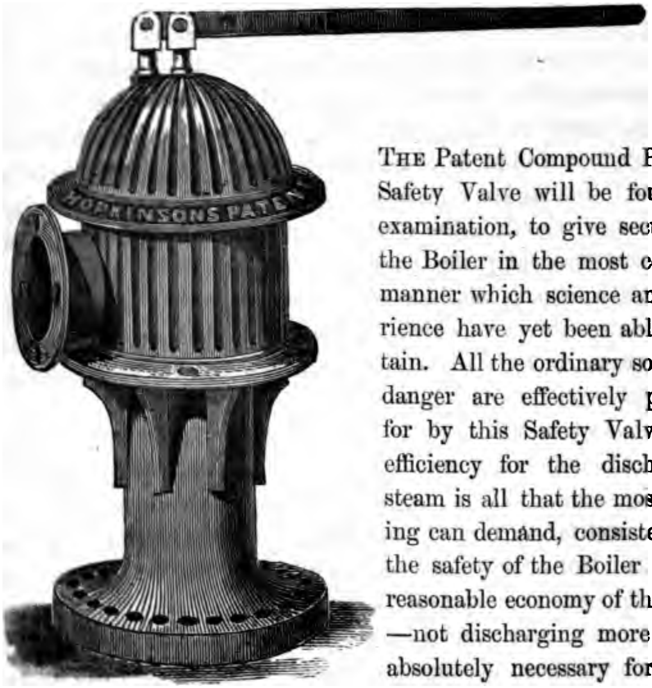
In all cases it is advisable to have the face of the seating of the Safety Valve as narrow as the varying modes of construction will allow. The Spherical-faced Valve is best adapted for this and other beneficial purposes: for it is found in practice that the sharp edge of the seating, taken off only by grinding it and the Spherical Valve to a face, results in the greatest accuracy that can be obtained—while this form and mode of construction also insure the best fitting valve for "bottling up" the steam when it is not required to escape.

The valve shown in the foregoing Diagram is not adduced and recommended as one perfect in every particular; but it is believed to be as good a valve as can be made simply as a Lever Valve—securing the greatest accuracy of which this form of construction is capable—one providing for the discharge of steam better than any other mere Lever Valve yet introduced.

In connection with this subject of Safety Valves, the Authors would not be doing their duty to those for whom this work is intended, were they, from any feeling of mock-modesty, to refrain from describing the construction and the advantages of what is now very well known as "HOPKINSON'S (PATENT) COMPOUND SAFETY VALVE"—as that valve possesses important features not to be found in connection with any other valve: and as it provides against that prolific source of Boiler explosions, deficiency of water, as well as for the copious discharge of steam at over-pressure, under the ordinary conditions of working. When the facts are stated, that this Compound Valve has been applied to upwards of fourteen thousand Boilers, and that in no single instance has a Boiler with this valve attached exploded or suffered injury, either from over-pressure or deficiency of water; while most numerous are the instances where Boilers have been *saved* by the simple, but certain, action of the instrument, under circumstances which, with the ordinary valve, would inevitably have resulted in explosion,—the Authors feel that they may fairly claim to be exempt from the charge of "puffing their own wares," in giving the

following description of a construction which has already effected much good, and which, as experience warrants them in saying, is the most effectual *Safety* Valve ever yet introduced into Steam Engineering.

THE PATENT COMPOUND PYRAMID SAFETY VALVE.



The Patent Compound Pyramid Safety Valve will be found, on examination, to give security to the Boiler in the most complete manner which science and experience have yet been able to obtain. All the ordinary sources of danger are effectively provided for by this Safety Valve. Its efficiency for the discharge of steam is all that the most exacting can demand, consistent with the safety of the Boiler and the reasonable economy of the steam,—not discharging more than is absolutely necessary for perfect safety.

It provides for low water in the Boiler, and secures the opening of the valves whenever the water may (from whatever cause) get down to a given point of lowness,—the valves opening more and more as the water still lowers, so that no dangerous pressure can possibly remain in the Boiler,—thus obviating all danger of explosion, which would otherwise be imminent. The valve furthermore provides for the discharge of steam whenever the water shall get too high. And these results are all attained by a valve which cannot by any possibility be interfered with when the Boiler is once filled with

water. A lever is placed outside, by which it can be tested, but which does not enable any person to interfere with its action. The valves are held down by a dead weight inside the Boiler, and are so shaped and arranged that there is no liability to sticking under any circumstances. The valves being multiple, are of smaller area than usual, although having a much greater discharging power. The smaller area is advantageous in two ways. Universal experience has proved that a valve of small area is not so liable to become leaky, as a valve of larger area; therefore the Pyramid Safety Valve must secure such advantage. In the next place, the comparatively small weight required to hold down the valves permits the valve seating to be made feather-edged, thus leaving small surface for dirt, or any other matter whatsoever, to rest upon, which could cause the valve to be leaky. The Compound Pyramid Safety Valve is the only one which secures all these conditions.

We will now give a more detailed description and exposition of it; and for this purpose we cannot do better than give the following copy of our descriptive circular.

"The Patent Compound Pyramid Safety Valve is truly what its name imports,—a means of safety under all circumstances which ordinarily produce Steam Boiler Explosions, viz., over-pressure of steam and shortness of water.

"The valve we now introduce is an improvement over every other description of Safety Valve hitherto adopted, and as it is imperative that every Boiler should be fitted with a Safety Valve, it behoves the owners of Steam Boilers to adopt such fittings of this character as are efficient for their purpose.

"A Select Committee of the House of Commons have sat to inquire into the cause of Steam Boiler Explosions, and the means of prevention. Evidence of a most searching and scientific character was given. It was shown that a Boiler requires careful examination previous to being set to work, and that during its working career it should receive a thoroughly practical periodical inspection from time to time. It has also been shown that although an examination is greatly to be desired, it merely serves to show that the Boiler is capable of working at a certain pressure of steam, under the proper conditions for safe working, and that for such conditions we have to look to the fitting up or mounting of such Boiler. The Safety Valve, *then, is admitted by all to* be the means, or at least one of th

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first means, by which we obtain security, and we purpose to examine and afterwards show what the Safety Valve is, and what it should accomplish.

"Having been engaged for twenty years in this branch of Engineering Science, and having applied to Boilers in various parts of the world *Fourteen Thousand Safety Valves,* we have now by recent improvements so perfected the Safety Valve, that Boiler explosions arising from over-pressure of steam and shortness of water need not occur.

"The essential requirement of a Safety Valve is to relieve a Boiler of its surplus pressure, over and above that to which it is weighted, but hitherto it has been found that valves could not be constructed to fully accomplish the object; and to approximate to anything like an adequate relief for any degree of safety, valves of large and unsafe sizes have been made, or otherwise several valves have been placed upon one Boiler. This arises from the fact that it is impossible to raise a valve from its seat by means of steam pressure more than just an imaginable separation of valve from valve seat; hence it is, that nearly the whole orifice of a valve is literally and practically useless as a means of discharging steam, unless sufficient circumferential area can be given in proportion, or equal to the area of the valve orifice. For instance, a 4 inches diameter valve possesses 12½ square inches area, and a circumference of 12½ square inches; it is therefore plain that to present a circumferential area or opening for discharge equal to that of the orifice on which it is seated, that the valve would have to rise exactly 1 inch high from its seat, and as the size, or rather diameter, of the valve increases, it would require to rise still higher from its seat, to give a proportionate discharge to the increased area, as in case of an 8 inch diameter valve, whose area is 50 square inches, whilst the circumference is only 25 inches, this would require to be raised exactly 2 inches from its seat, in order to give an area, or power to discharge steam, equal to the area on which the valve is seated, and this is *totally* impossible, for when a valve is discharging steam furiously for over pressure, it is merely a separation of valve from valve seat.

"Thus, for example, if a valve of 6 inches diameter could lift $\frac{1}{32}$ part of an inch from its seat, it would merely give an area for discharge $\frac{1}{32}$ *of an inch* on the circumferential measurement of the valve,

which will be $18\frac{3}{4}$, and half of a square inch, leaving $5\frac{29}{32}$ inches diameter, equal to $27\frac{3}{4}$ inches area, completely useless as a Safety Valve, or discharge for steam; the whole area of the valve orifice is 28·274, or a little over $28\frac{1}{4}$ inches area, whilst an area of 27·684 inches has to be weighted uselessly to obtain merely a discharge from its circumference of 0·59th of an inch, or nearly $\frac{19}{32}$, so that it will be seen that its power to discharge steam is but $\frac{1}{48}$th part of the whole orifice; or, in other words, 48 times more weight than is requisite is employed to weight the discharging power. This amount of area is of no benefit whatever. If worked at a pressure say of 80lbs per square inch, it would require 2,215lbs weight, whereas the total weight required is 2,262lbs, leaving only 47lbs of that weight available to weight the discharging power: the 2,215lbs having to be employed to weight a great diameter of orifice, simply to obtain a single circumference for discharge.

"Seeing then it is but just the edge of a valve that really gives the power or capability of its discharge, and that as we doubly increase the diameter of valve, we increase the area to be weighted fourfold, and its power to discharge but twofold, it is obvious to all, that to make a valve powerful, is to make available as much of the orifice as possible, or in other words, to give a power or area in the circumference, equal or approximating to the area of the orifice, and this is one of the objects of our invention, as we will now proceed to describe.

"DESCRIPTION.

"On a seat of any given diameter, we place a valve of the ball shape on a feather edge seat; this valve forms the seat for another valve placed on it, slightly less in area, also of the ball construction, and seated on a feather edge as before; on the second valve is placed a third one, again slightly less in area, and seated as before, and on the third valve is placed a fourth valve, again slightly less in area, and seated as before; to this valve is suspended a dead weight equal to the area presented for the pressure required; and, by placing a pile of valves one upon another, and varying the area as explained, four separate and distinct discharges are obtained, of equal circumference to four separate and distinct valves, yet the amount of weight or holding down power is what would be required for a single valve only of the same diameter.

"These valves placed one above another in the Pyramidal form

may be carried out to any number, until their aggregate circumferences equal the capacity or area of the orifice, and each and all act separately and distinctly as would a single solid valve.

"This mode of seating and weighting valves may be varied according to circumstances. We will describe the broad feature or principle of our invention, and that it may be more easily understood we will proceed to make a comparison with Safety Valves of the ordinary dead weight and single discharge construction. Take, for instance, one of our Improved Valves for 40lbs pressure per square inch, the area of which is 7 square inches, whilst its combined circumference or power of discharge is 36 inches, which to mark at a pressure of 40lbs would require 280lbs of dead weight suspending to it, but in the case of an ordinary dead weight and single discharge Safety Valve such as usually used, it would take a valve 11½ inches diameter to possess the same circumference or power of discharge, and to weight it at 40lbs pressure would require 4,146lbs, as against 280lbs in the new Pyramid Valve; or, for further example, take the case of one of our Improved Pyramid Valves for 80lbs pressure, the area of which valve is 4 inches, and its circumference or power to discharge is 29 inches, the weight required is 320lbs. An ordinary valve, to possess the same discharging power, would require 9¼ inches diameter, and have suspended to it 5,376lbs weight; there being also a great risk with large and heavily weighted valves, and a difficulty in making them efficient for the discharge of over-pressure. It is clearly an error of no small magnitude employing heavily weighted single discharge valves when so many advantages are gained in the New Pyramid Valve.

"Another feature of our Improved Valve is the sensitiveness and the freedom of each part rising from its seat in consequence of the narrow face and the little weight suspended to them in comparison to other valves of equal circumference, and from the multiplicity of its seatings, which are all feather edged, and the absence of joints, levers, &c., &c., there is no possibility of its sticking and becoming deranged. This then serves to show that the diameter of a valve is not necessarily a criterion of its discharging power, but the greatest circumference with the least weight is the great desideratum, and that the true object is to make available the power of a small orifice, rather than make valves of large diameter and get no advantage whatever.

"Besides possessing the capacity of discharging all the surplus steam which a Boiler will generate, and rendering it secure against over-pressure, provision is also made for shortness of water, which, perhaps, is an equally important feature of our invention; for this purpose, we have an ingenious arrangement of float submerged or placed below the surface of water at such a height as is deemed low water mark. Should the water become by any cause below a certain fixed level, the float raises the valves just in proportion as the water lowers, and gives timely and unmistakable notice that danger is approaching, which; if neglected, the valves will inevitably open to the full extent, letting off the steam and preventing all risk of explosion.

"The advantages of the New Patent Pyramid Safety Valve are as follow:—its combination of parts, which are such as act for excessive pressure and deficiency of water, its general mechanical and practical arrangements, the valve possessing neither guides, spindles, rubbing surfaces, nor complicated parts, &c., liable to adhere. It is simple in construction and certain in action; while it can be used as any other valve for general working, it prevents the careless, the ignorant, or the wanton, from causing either injury to the Boiler, or Boiler explosion, it is not liable to derangement, and is in every detail what a Safety Valve ought to be.

"Having now shown the means by which the serious catastrophes of Boiler explosions can be prevented, viz., by ascertaining the capability of the Boiler to withstand its working pressure, and the adoption of the Patent Compound Pyramid Safety Valve above described, it rests with the owners whether such calamities are to continue to occur. We ask, then, of all Boiler owners and users of steam power the question, that (asserting these statements as facts which cannot be gainsaid) are they not acting prejudicially to their own interests, to the jeapordy of their own and workpeoples' lives, as well as to the durability of Steam Boilers, to continue to work and put down Steam Boilers devoid of this Patent Valve, when such can be applied for *less money* than adopting the old, unsafe, and obsolete fittings?

"Deficiency of water, although often not actually the proximate cause of an explosion, is the primary one in many cases, for injures a Boiler, and is costly to the owner; the adoptior this adjunct *dispenses with* such risk and liability, and rer

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from which the ball weight is suspended. Divide this by the former, and the quotient is the number of times which a given weight of ball will press on the valve or the lever ; ascertain the area of the valve in square inches, and multiply this by the number of pounds pressure per inch at which the steam will blow oft. This will give the total pressure exerted by the steam on the valve. Then divide this by the leverage, and the quotient will be the weight in pounds of the ball weight.

EXAMPLE I.

Let the area of the Safety Valve be 12 inches = 3·909 inches diameter. Let the length of the lever be 40 inches from the fulcrum* to the point on which the ball weight rests, and the length from the fulcrum to the point which rests on centre of the valve 4 inches. Let the blow-off pressure be 60lbs per square inch. Then, as the length of the lever to the valve is 4 inches, and to the ball weight 40 inches, so one pound weight at the latter place is equal to 10lbs on the valve. The area of the valve orifice being 12 inches, and the pressure of steam 60lbs per inch, the total pressure of steam on the valve will be $12 \times 60 = 720$lbs. Thus, as one pound weight at 40 inches will equal 10lbs on the valve, so 720lbs $\div 10 = 72$lbs the weight of the ball.

EXAMPLE II.

Area of valve, 12 inches	Lever = 4 inches to valve
Pressure ℔ in. 60lbs	Lever = 40 inches to ball weight
720	

Therefore, as 40 inches is to 4 inches, so is 720lbs to the weight required.

Thus—40 : 720 : 4

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          4
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40 ) 2880 ( 72lbs on lever at 40 inches
     280
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      80
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In the above calculation the weight of the lever has not been taken into account, but this must be provided for in *one* of *two* ways. First, it may be made to extend past the fulcrum, on the other side to that of the valve and ball weight, a distance sufficient to fix on it a weight which will simply balance the lever on the fulcrum, so that no weight will rest on the Safety Valve before the ball weight for the calculated pressure is put on.

Second—Another way of providing for the weight of the lever is this: Place the lever in a horizontal position, with the fulcrum pin

* The fulcrum is the centre of rotation of the lever, and the measurement must be taken from this *centre*.

in its place, and ascertain the weight or pressure at the point of the lever which rests on the centre of the Safety Valve. This may be done by scales, in a variety of ways.

Then, whatever may be the weight of the lever on the Safety Valve thus obtained, it must be deducted from the total pressure of the steam on the valve, and the remainder will be the pressure to be provided for by the ball weight. For example—Take the case already calculated, where the total pressure of steam on the valve is 720lbs; we will assume that the lever, when tested as just described, gives a weight or pressure on the valve of 60lbs, which being deducted from 720lbs leaves 660lbs to be supplied by the ball weight, and as this gives a pressure on the valve of ten times its own weight, so 660lbs÷10=66lbs required.

The introduction of the above simple method of calculaton is not for the purpose of superseding any readier method, or those usually adopted by the mathematician, and the skilled and practical Engineer. It is intended for those who have not any other means of arriving at the required result.

CHAPTER II.

THE FURNACE.

PERHAPS there is no more important part of the question of Steam Engine economy than that of the arrangement and condition of the Furnace; and yet it has been too often neglected, or not understood.

The subject embraces the sciences of chemistry and mechanics—equally important, and closely allied in their bearings on the question; therefore, furnaces should be constructed with a view to the application of the laws of these sciences in the combustion of fuel and the utilization of the greatest amount of heat attainable. The Furnace has received little material improvement since its introduction; yet, there is no part in connection with the Steam Engine for which so many patents have been obtained, and so many ideas re-patented. There are so many circumstances and conditions which affect the Furnace in different situations, that it would be impossible to lay down any precise rule for its construction in every case. Still there are certain known principles, with reference to the combustion of coal, which cannot be ignored, and therefore should never be lost sight of.

Chemically speaking, all Furnaces should be constructed to consume any given quantity of coal with the greatest possible amount of heat; and, mechanically speaking, it should be applied to such portions of the boiler as will be the most effective in the generation of steam. But it is first necessary, and absolutely so, that a good draught be obtained. No matter how well a Furnace be designed, if there be not a good draught its economic value will be impaired. In the first place, there must be a good chimney to have a good draught. It is not necessary to have a very high one; but it is all-important that it should be properly proportioned. Though the chimney may be such as would ensure a good draught, yet, if the flues which surround the boiler be not of suitable form and proportion, the draught will still be defective.

We have seen some hundreds of cases where the draught in the chimney has been excellent, whilst the Furnace has been little more than a retort, generating gas from the coal, but a great portion of it never created into flame and heat. Now, good draughts are spoiled by many causes; some are rendered defective by being crippled, others by having some very quick corners to turn. And it is well known to practical men that a draught cannot be more seriously injured than by bringing two or more currents in direct collision with each other. Or, it may be injured by having crevices in different parts of the flues, through which the air may enter. Dirty flues, which have the effect of contracting the space through which the gases must necessarily pass, also injure, to a considerable extent, the draught. No money is more wisely spent by the proprietor than in keeping the flues clean; for a new or clean boiler will generate a given quantity of steam with less fuel. It is, therefore, important that this should be kept in mind.

Suppose now that a good draught has been secured in the Furnace, which is the first essential in the economical combustion of coal, how must we proceed to work this Furnace that every pound of coal shall do its duty, or, in other words, give out the greatest amount of heat.

To effect this, the greater the quantity of coal which can be consumed per square foot of grate surface, the better will be the result, and the greater the economic effect. The late Mr. HOPKINSON, when contesting for the £500 prize, offered by the Newcastle Coal-owners' Association, some years ago, found, that when he had the greatest heat in the chimney, he obtained the best results. This may be surprising to some, yet, it is nevertheless a fact, and has been proved many times since. Indeed, it is a conclusion borne out by common experience.

Take, for instance, a Furnace with a grate surface of 20 square feet, burning 12lbs of coal per square foot per hour, with a temperature of 1,500°, and a temperature in the chimney of 500°, which would be a fair and reasonable temperature to assume under these conditions. Let the grate of this furnace be now altered to 15 square feet, and burning 16lbs coal per square foot of grate per hour, and the results would be quite different. Instead of having a temperature of 1,500 degrees in the Furnace, as in the former case, there *would be one of* something like 1,800° or 2,000°, with a corresponding

increase in the chimney. This increased temperature, generated by the same quantity of fuel, acts correspondingly on all the heating surface of the boiler, and certainly increases the economic value of the fuel.

Another proof of the importance of a proper proportion of fire-grate area is supplied by the following case :—An extensive firm had a set of nine boilers externally fired, with a consumption of coal of 17 tons per day. The grate surface for each boiler was $6 \times 5 = 30$ square feet, which, on being reduced to $3 \times 5 = 15$ feet each, the consumption of coal was reduced to 13 tons, by the aid of judicious stoking. These, with many more cases which could be cited, show the fallacy of having a long and large area of grate. It is not only very inconvenient to work, the back part being often bare, but is a great loss in other respects. It is better to have the bars so short that it will require the full power of the draught, when the coal will burn with a more intense heat, and therefore with greater economy.

When coal is applied to a Furnace, the first action which takes place is the liberation of the hydrogen and hydro-carbon gases ; but if these gases are not consumed when generated by a proper admixture of oxygen, they pass away and are lost ; whereas, with proper conditions, they unite and produce heat in the highest degree, and may thus ensure the greatest evaporative efficiency.

The quantity of air admitted should be such as will suffice for the most perfect economical combustion of the fuel ; but if in excess of this amount, then the temperature of the resulting gases will be lowered, and, as an inevitable consequence, fuel will be wasted.

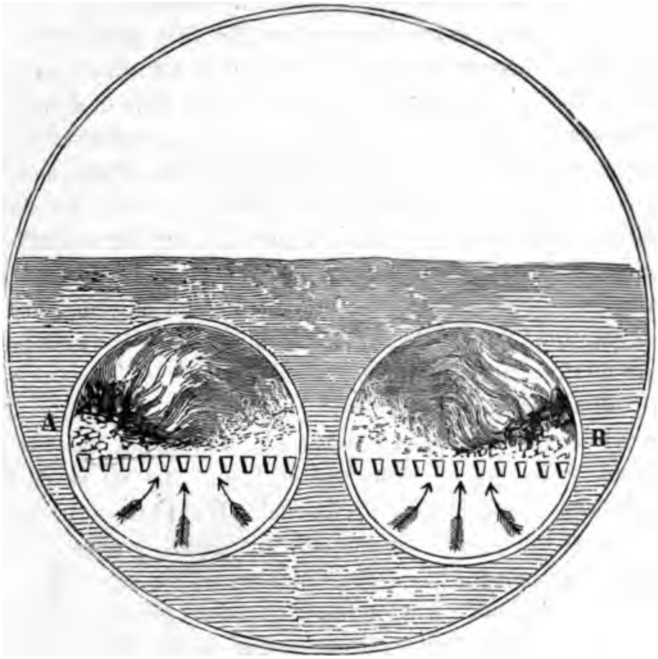
In the next place, the nature of the coal should be ascertained—whether or no it be of a coking description. If it be, the grate bars will require to be more open than for coal of a light and gaseous nature. Adjust the grate-surface to the heating or absorbing surface of the boiler. This will greatly depend upon the quality of the fuel and the draught, and the quantity of the steam required. If the draught be good, work a thick fire ; but do not break the coal except it cannot be got in at the Furnace door, taking care to feed the Furnace on one side, not covering more than one-half the grate at a time. When that is sufficiently coked, feed the other side, putting plenty on at a time, leaving the middle of the Furnace much lighter and thinner of fuel than the sides, and keeping the grate bars well open. *This mode of* stoking—placing the coal on one side

at a time will prevent the formation of a great portion of the smoke which would otherwise result, and also produce a more steady heat. The coal so placed will partially damp that side of the Furnace on which it is placed; *therefore the necessity of stoking the Furnace when the steam is up.* From this method of firing, the gas generated will be given out slowly. The greatest quantity of air will pass through that portion of the grate where the fuel is most consumed, and where the coal has not been made small by breaking, leaving the interstices wide; and as the oxygen of the air will generally be in sufficient quantity, if this mode of firing be adopted, to unite with the gas as it is extracted from the coal, a continuous flame will be kept up, and smoke prevented. When one side of the Furnace so fired is sufficiently coked, raise the burning fuel gently with the poker, but do not break it into pieces. At the next firing, place the coal on the other side of the Furnace, and similar results will follow. Keep the Furnace door open no longer than is absolutely necessary, and thus prevent the admission of large volumes of cold air to cool the boiler and flues. Place your coal, before firing, near to the Furnace, that there may be no loss of time in charging the fire. Cold air coming in contact with the boiler lets down the temperature, and does harm. To ensure that the coals be sufficiently near, and that the Stoker cannot have them too far from the Furnace doors, it is best to erect a barrier in front of the boiler, at about six feet distance, with an opening at the bottom. The coals behind this barrier will have to be put up to the opening at the bottom. This mode will enable the Stoker to charge his Furnace with fuel without moving from the front. He will thereby take less time, and be able to close the door earlier, than if he followed the ordinary modes of stoking.

If the Engineer is working a Double Fire-box Boiler, let him fire each Furnace alternately, in the manner described in Diagram No. 9, where A and B represent the sides of the fires last stoked, with the gas given out from the new fuel, mixing with the air which passes in the greatest quantity through the thin portion of the fire, as indicated by the arrows: and with that also which passes through the apertures in the Furnace doors, or through any special provision that may be made. Being thus mixed with air at a high temperature, the gases ignite, and comparatively thorough combustion is secured, preventing smoke and saving fuel. This mode of firing is applicable

to most Furnaces, whether single or double fire-box, or with the fire wholly beneath the boiler. The hotter the fire, the more steam will there be generated with the same quantity of fuel; and though a greater amount of heat may pass up the chimney, there will be a corresponding saving of fuel, in proportion to the increase of temperature.

Diagram No. 9.



There are various qualities of coal, and various power of draughts. A good quality of coal, with a good draught, will evaporate 7lbs of water for each 1lb of coal, in a Double Fire-box Boiler; and other qualities not so good, from 5lbs. to 7lbs. In a trial at Newcastle-upon-Tyne, Mr. HOPKINSON evaporated 11·70lbs of water with 1lb of coal, without any smoke being evolved. The experiment was made with a Marine Boiler, with multitubular flues. The temperature in the chimney varied from 500° to 650°; and it was found that as the temperature of the Furnace was increased, the heat in the chimney also increased: but steam was generated much quicker, and less fu was proportionately *consumed*.

If the coal be of a light gaseous description, burning to white ashes, work the Furnace in the manner above described, with the bars a little closer together. When there is a bad draught, the Furnace will require working differently—placing a little coal on at each time of firing, equally over the grate. But this mode of firing is, generally speaking, not an economical one. Much, however, depends on the nature and quality of the coal—for there are some descriptions which will not fire in any other manner. By this latter mode of firing, the gas is extracted from the fuel quicker than the oxygen passes through the grate to consume it. In that case a great portion of the fuel passes up the chimney in the form of smoke; and the Furnace door having to be often opened to replenish the fire, admits large quantities of cold air, which cools the Furnace and the flues, besides contracting the boiler—an evil somewhat modified by partially closing the damper when firing. A dirty Furnace will have a similar effect in creating smoke. If the grate be covered with clinkers or rubbish, the air is prevented from entering, and the fuel is converted into smoke. In fact, this is one of the most prolific sources of the "smoke nuisance."

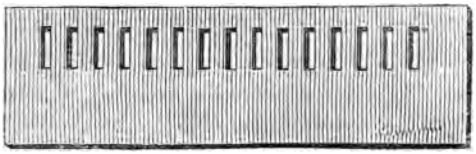
By all means avoid having a dirty ashpit. A pit nearly full of ashes is a sure sign of bad management. It is as necessary to keep the ashpit of a Furnace clean and cool, and the air as dense as possible, as it is to feed the fire with fuel. If the ash-pit be hot, the heat expands the air, and the Furnace will require *greater quantities* to pass for the amount of oxygen which would be supplied by less quantities of cold air. These greater quantities impair the draught. When convenient, use water for the bottom of the ashpit, allowing it to remain to quench the ashes as they fall from the Furnace. The evaporation will tend to keep the grate cool. Have, if possible, some water near the Furnace, so that when cleansing the fire you can cool the slag, that you may remove it early. These things, properly attended to, will cause the Furnace to wear much longer, and also save fuel. There will likewise be less labour to produce more steam in the same time than a slovenly mode of stoking would produce.

A great body of fire will evaporate more water with a less proportion of fuel than a small and thin fire. The greater the intensity of the fire, the more will be the steam generated with the same amount of fuel. Where there are a number of boilers, it is impor*tant that the duty of* stoking should be properly attended to; and as

smoke is a great nuisance, its prevention is a duty which every Engineer ought to attend to.

With regard to the consumption of smoke, it may be observed that the heat of one fire coming into contact with the gases of another fire behind the bridge, or in what is called the " combustion chamber," does not consume the smoke. Smoke cannot be consumed after it has left the Furnace, unless air be admitted to mix and ignite with it, and the gas must also be of sufficient temperature to cause ignition. The best part of the Furnace in which to admit air to ignite the gas, is near to where the gas is generated. If additional air be required to that which passes through the grate, it is best admitted through the dead-plate, as illustrated by diagram No. 10, which represents a dead-plate pierced with apertures for the

Diagram No. 10.



admission of air. This method of admitting air has been much resorted to since the first edition of the work in which it was described and made public, first appeared.* This method is recommended by the " Lancashire Association for the Prevention of Boiler Explosions ;" and Mr. HOPKINSON, having had more than twenty years' experience with almost every kind of smoke-prevention apparatus, is able to say that he never found anything to answer so effectually as this mode of admitting air through the dead-plate, when the requisite amount cannot be got through the ordinary grate.

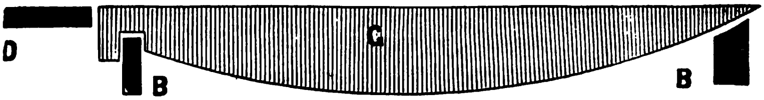
All Furnace doors should have a number of small holes cast or drilled through, independent of all other contrivances. The air passing through these holes into the Furnace keeps the doors cool, and assists combustion.

The annexed Diagram, No. 11, is a representation of a grate bar, and bearers. The bar C shows one end hooked on to the left hand bearer B. The other end of the bar slopes, and rests upon the other or right-hand bearer B. As the bar expands or contracts, the

* " The Steam *Engine Explained by the use of the Indicator*," by Joseph Hopkinson.

sloping end, being loose, can move without straining, or buckling the bar. With this free action, the grate will last much longer than ordinarily, and the openings can be kept more even.

Diagram No. 11.



The form of the Furnace bridge is also of importance. If it be made inclined, with the back part highest, it will be much better, and the heat will be more evenly distributed in the Furnace, to the increase of the evaporative power. When the Furnace is cleansed, the hot fire can be pushed up the incline, while the clinkers are removed,—thus enabling the Stoker to clean the whole of the grate at one time. And, besides this, the reflected heat of the hot brick-work will impinge against the top part of the fire-box flue, or the bottom of the boiler, if of another construction; thus keeping the Furnace-door much cooler than when the bridge is built perpendicularly—the reflected heat in the latter case impinging on the front, causing that part to be much hotter than where the bridge is inclined, as here described.

The opening between the grate-bars is an important point to be attended to. If these are not sufficiently wide, the Furnace becomes a retort for the destructive distillation of coal, the gases passing off in the form of smoke for want of a sufficient quantity of air to consume them as they are generated. Smoke is often produced from the bars being too close together; and it is false economy to suppose that fuel is saved by having the apertures between the grate-bars too small. A proper thickness of the bars, also, is a requisite in practice. These have been found to answer best when about three-quarters of an inch thick.

Attention to the foregoing rules, viz.: careful firing, clean grates, wetting the small coal, placing plenty on at a time, firing alternately the sides, a clean ash-pit, and, above all, a good draught, will, for all practical purposes, prevent the formation of smoke. That attention *will also* prevent waste of fuel, save time and labour, and reduce *wear and tear*. Of course, under different circumstances, modifica-

tions will be required, varying accordingly; but by the adoption of the principles herein laid down, the "smoke nuisance" would soon cease to be a cause of complaint.

The following table of the evaporative powers of various descriptions of coal when burned under boilers, is given by Mr. Wicksteed:

No.	Description of Coal.	Water evaporat'd per ℔ of Coals	Comparative Cost per Ton in London	
		lbs.	*s.*	*d.*
1	The best Welsh	9·403	17	11
2	Anthracite	9·014	17	0
3	The best small Newcastle	8·524	16	1
4	Average small Newcastle	8·074	15	2¾
5	Average Welsh	8·045	15	2¼
6	Coke from gas works	7·908	14	11
7	Coke and Newcastle, small, half and half	7·897	14	10¾
8	Welsh and Newcastle, mixed, ditto	7·865	14	10
9	Derbyshire and small Newcastle ditto	7·710	14	6½
10	Average large Newcastle	7·658	14	5⅛
11	Derbyshire	6·772	12	9¼
12	Blythe Main, Northumberland	6·600	12	5⅛

Mr. Hopkinson evaporated 11·70lbs of water in a Marine Boiler, with 1lb of the best Newcastle coal, when contesting for the £500 prize offered by the Newcastle Coal Owners' Association some years ago, for the best smoke preventive; and he further offered to evaporate 12·70lbs of water with 1lb of coal, provided the Furnace was altered as he might direct.

With good Lancashire or Yorkshire coal, and a Double Fire-box Boiler, new, and a moderate draught, 7lbs of water can be evaporated with 1lb of coal. When the boiler has been some years at work, the evaporation will vary from 5lbs to 7lbs — the work greatly depending upon the state of the flues and the cleanliness of the boiler-plates, inside and outside. Inferior coal will not evaporate more than from 4lbs to 5lbs of water with 1lb of coal. The strength of the draught has a good deal to do with the economy of fuel: the stronger the draught, and the greater the evaporative power of the coal. The greater the intensity of the fire, the more completely are *the gases consumed.*

CHAPTER III.

THE ENGINE.

THE Engine itself requires but little attention when in good order, except the ordinary routine of packing, cleaning, and oiling the working parts, and occasionally adjusting the cotters, which on no account ought to be neglected.

Indicate the Engine every day. Perform the operation when the load is on. Work up the diagram, and place it in a book, with the date, for reference at any future time. This mode will denote the difference of power required under different circumstances, and also make known any defective working, whenever any part of the Engine gets out of order, or whenever any shafting or machinery is taking more power at one time than another.

It will not be necessary to work up the diagram each time. When the indicated horse-power has been once ascertained, the following rule will be found to be correct enough for all practical purposes, and much more ready than working the diagram each time thoroughly out.

EXAMPLE.

Say the diagram represents 99 horse power, when the friction of the Engine and shafting is deducted.

Divide by the average pressure,
say...12lbs) 99 (8¼lbs horse power for each pound pressure
96 ℔ square inch upon the piston
—
3 remains ; this being ¼ of 12, it makes the total to be 8¼ horse power

If a diagram be taken from the same Engine, with less or more average pressure, multiply the average pressure upon the piston by 8·25, to ascertain the horse power the Engine is then exerting.

EXAMPLE.

15lbs the pressure upon the piston
8·25 the number of horse power for each pound pressure upon the piston
——
75
30
120
——
123·75 horse power.

The horse power being 123¾lbs. It matters not whether the horse power be required independent of the shafting and Engine, the calculations are made the same. Each Engine indicated will require the first diagram taken to be figured up, and worked out in the ordinary way (hereinafter described), to ascertain the horse power the Engine is turning.

Use a thermometer to test the temperature of the condensing water, which ought not to exceed 120° Fahrenheit. Work as much below that temperature as is consistent with the supply of water to the boiler, and the quantity the air-pump will discharge—keeping the delivering-valve closed as long as possible. If the water in the air-pump thumps and plunges, a small air-pipe inserted in the pump lid will often prove a remedy.

Unsteady working of the Engine is sometimes caused by the connections between the governor and the throttle-valve being too weak; in that case the parts spring and give way, and vibration is the consequence. When the throttle-valve is too large, a similar effect is produced; and this is often the cause of Engines working unsteadily. Throttle-valves regulated by pumps, vary with the pressure of the atmosphere; for the water is "high" or "low" in accordance with the varying pressure.

Should the crank-pin, or the bearings of the fly-wheel shaft, or any of the other parts become hot, particular attention is required to prevent their destruction. Lead filings mixed with oil will be found to be useful. The lead coats the bearings, and interposes another body between the rubbing surfaces. Sulphur mixed with oil has also a similar effect; and quicksilver may be said to be better than either.

The following recipe for cooling necks of shafts will be found useful, not only for the crank shaft, but for all other descriptions of shafting—particularly the feet of upright shafting:—

16lbs of Tallow, dissolved in a vessel. 2¼lbs of White Sugar of Lead. When the tallow is melted, but not boiling, put in the Sugar of Lead, and let it be dissolved. Then put in 3lbs of black Antimony. Keep stirring the whole mass till cold.

It is an axiom that "prevention is better than cure." There would seldom be heating of necks of shafting, were those necks made of a proper length. They should, in all cases, be at least *twice* the length *of the diameter of the* shaft. The "brasses," or "steps

should be made of good metal, composed of three parts of copper to one of tin. Good brasses involve more "first cost," but they secure a saving of oil, and also of power. More attention than is now usual ought to be paid to the quality of the metal of which "steps," or the bearings of shafting, are made. Above all, be careful to keep dust away from the necks and brasses of shafting, and other bearers, for cleanliness in this particular is a great preventive of destruction.

Pay proper attention to all packings, to prevent the escape of steam or water. Such leakages often cause more destruction than ordinary wear and tear. If the injection cock or the condenser be leaky, the water will get into the cylinder when the Engine is standing, in consequence of the partial vacuum; and at starting afterwards, the water will, in all probability, be the means of breaking the beam or some other part. Care should be taken that all taps connected with the cylinder are closed when the Engine is at rest, except a small tap connected to the condenser; and whenever the Engine is stopped, this tap should be opened to destroy the vacuum, or the water may, from some cause, get into the cylinder. At starting the Engine, close the tap. This simple arrangement will be the means of saving much trouble, and probably prevent breaks-down. This tap will also be found useful in stopping the Engine at any required angle of the crank, by admitting air to the condenser, and destroying the vacuum.

Keep the pipes and cylinders clothed with a non-conducting substance, to prevent the escape of caloric from the steam. It will pass away very quickly, if means are not used to retain it,—causing a loss of fuel.

Have a vacuum-gauge fixed in the Engine-house, near to where you pass, that the quality of the vacuum in the condenser, or the amount of the uncondensed steam may be easily examined. The vacuum-gauge should be one that shows the pressure of the uncondensed steam left in the cylinder, and the pressure of the atmosphere at the same time. Otherwise it cannot be ascertained by the *Indicator*, when the vacuum varies, whether it be from the pressure of the atmosphere or the deficient vacuum of the engine. In some states of the weather the pressure of the atmosphere will be under 14lbs to the square inch, and at other times it will be upwards of 15lbs to the square inch. Steam may be 20lbs above the atmosphere; but when the pressure of the latter is 14lbs, the steam

will be 34lbs. But if the pressure of the atmosphere be 15lbs, to the square inch, the steam in the boiler will be 35lbs; and so on in proportion to the varying pressures of the surrounding medium.

Have proper taps fitted to the top and bottom of your cylinder, that you may indicate your Engine with ease. By indicating each end, you will see if your valves are equally set. Both ends can be indicated on one paper. After using the *Indicator*, clean it well: a dirty *Indicator* is an indication of a slovenly Engineer.

WHAT IS THE VACUUM OF A STEAM ENGINE?

IT is essential that we define and understand the term "VACUUM," as commonly applied to the Steam Engine—(and unless this be well understood by the Engineer, he cannot clearly understand the Condensing Engine)—before we enter into a description of the diagrams which are hereafter given. The following popular explanation is therefore attempted.

Steam is an invisible fluid. When steam, as it is called, is seen like white smoke, it is not steam, but minute particles of water. Pure steam in a glass will not show at all—but the vessel appears to be filled with air only. One cubic inch of water converted into steam at the atmospheric pressure of 14·7lbs, the temperature 212°, will expand to 1700 cubic inches of steam. If the temperature be increased, the pressure will be increased also in the ratio of the temperature. Thus, with steam at 250·4 degrees, the pressure will be 15lbs above atmospheric pressure; the total pressure, 30lbs to the square inch, and the bulk 881 times greater than the bulk of the water from which the steam was produced. If the total pressure be 60lbs, or 45lbs on the steam gauge, the bulk will be 467 times greater than the waters the steam had been produced from. In proportion to the increase of pressure, so is the density.*

The literal meaning of the term "vacuum," is space unoccupied by matter. The cylinder of a Steam Engine filled with steam, though vaporised from a small quantity of water, cannot be said to be void of matter; but condense that steam to its original bulk into water, and withdraw this water from the cylinder, and the space

* See the *Table in Appendix, showing the* "Temperature of Steam at different pressures."

formerly occupied by the steam will be unoccupied. No matter remaining in the cylinder, there is what is termed a "vacuum," or a void space. We have supposed this operation to have taken place under the piston of a Steam Engine, and in that case there is no resistance to be overcome in the descent of the piston. The pressure of the atmosphere alone, which is 15lbs to the square inch, or thereabouts, would suffice to force the piston down with a power equal to the degree of vacuum formed, up to the limit stated—15lbs, if the vacuum be perfect. On this principle the first Steam and Atmospheric Engine was constructed: a cylinder with its upper end open to the atmosphere; steam was admitted below the piston to raise it, and this steam being condensed in the cylinder itself by the application of cold water, the pressure of the atmosphere alone caused the downward stroke in the manner above described. If steam be allowed to take the place of the atmosphere, as in WATT'S Engine, steam at atmospheric pressure will produce the same effect. 1700 cubic inches of steam, or one cubic inch of water converted into steam at atmospheric pressure, will have a force sufficient to raise a weight of one ton one foot high; two cubic inches of water, two tons; and each ton extra, one additional inch of water converted into steam. One of WATT'S first improvements was to attach to the Steam Engine a second vessel, in which to condense the steam. This he called a "condenser." He also introduced other alterations by which the vacuum was much improved, and the steam made to answer two purposes. First, by closing the cylinder with a cover, and admitting the steam between the cover and the piston, and connecting each side of the piston with the second vessel, or condenser, and the Engine cylinder, the Engine was thereby made to be double-acting, and its power increased in proportion to the pressure of the steam above the pressure of the atmosphere.

Steam arising from an open vessel—for instance, from the man-hole of a Steam Boiler—has a force greater than the pressure of the atmosphere, inasmuch as it has to displace the atmosphere before it can rise above the surface of the water. The resistance of the atmosphere is equal to about 15lbs on the square inch. It varies from 13¾lbs to upwards of 14·7lbs; on the average it is about 14¾lbs to the square inch. Therefore, steam enclosed in a Steam Boiler at 5lbs pressure per square inch above the atmosphere, or, in other words, at 5lbs on the steam gauge, is, in reality, a pressure of 20lbs

on the square inch, as applied to the piston of a Steam Engine under the conditions above stated—taking the pressure of the atmosphere at 15lbs. If the pressure be less, as it often is, say 14lbs, then the pressure upon the piston would be 19lbs, because the resistance of the atmosphere on the safety-valve and steam-gauge would be less, and the steam in the boiler also less, in proportion to the reduced pressure of the atmosphere. Hence it arises that an Engine heavily loaded varies in its speed with the varying pressure of the atmosphere. Suppose that the vacuum is not perfect—and in practice it never is so—and that there remains in the cylinder a portion of uncondensed steam, the resistance of which is equal to 3lbs to the square inch, then the steam on the upper side of the piston at 5lbs to the square inch above the pressure of the atmosphere, would act with an effective force of 17lbs upon the square inch: the upper side of the piston having exerted upon it a pressure equal to 20lbs to the square inch, and the under side a pressure, or resistance, equal to 3lbs to the square inch. Under these circumstances, the condenser will have exhausted steam from the cylinder equal to 12lbs to the square inch, commonly termed a 12lbs vacuum; and uncondensed steam will have been left in the cylinder, having a resisting force equal to 3lbs to the square inch. In proportion to the quantity of steam condensed to the whole, so is the value or available pressure upon the piston. If the uncondensed steam left in the cylinder were equal to 6lbs to the square inch, then, in the other circumstances supposed, the available pressure upon the piston would be only 14lbs to the square inch—a proof that vacuum is not power, as many are led to suppose. All power in the Steam Engine is derived from the pressure of the steam upon the piston. If there be no resistance on the other side of the piston, the whole pressure is available; when there is resistance, whatever be the amount, it has to be deducted. The available power of steam on the piston is what is left of the whole force when that deduction is made. The term "suction" is often used. There is no such process in nature, or in mechanics; and the use of the term only tends to confound the practical worker. All power derived from air, steam, or gas, is the result of pressure, or density; and in proportion to the pressure, so is the power.

If 5lbs more pressure to the square inch be added to the 5lbs before described, the pressure of the steam above the atmosphere will be 10lbs to *the square inch,* making only 25lbs as the available

[illegible] suppose that by doubling the pressure above the [illegible] pressure on the steam gauge, they double [illegible] steam, and that they also double the power of the [illegible] the case. They have only, in the case last [illegible] pressure of steam in the boiler above the [illegible]—added 5lbs to the 20lbs already available. [illegible] lbs to the square inch upon the piston of the [illegible] 20lbs. The increased power of the Engine is as [illegible] the non-resistance, or in other words, the [illegible] same. In the practical working of Engines, it is [illegible] pressure in the boiler can be brought to bear [illegible] To ascertain the real pressure operating on the [illegible] time, the *Indicator* has to be resorted to in the [illegible] explained.

[illegible] have reference only to Condensing Engines. [illegible] Non-condensing Engines, are constructed upon a [illegible] In them, the power of the Engine is as the [illegible] steam above atmospheric pressure. Steam at 30lbs [illegible] above the atmosphere, that is 30lbs on the steam-[illegible] the piston of a High-pressure or Non-condensing [illegible] force equal to the pressure in the boiler above [illegible] provided there be sufficient room in the passages [illegible] and the boiler, so that obstructions in the steam [illegible] the pressure before entering the cylinder—the [illegible] being open to the atmosphere, and the [illegible] overcome the atmospheric pressure in its escape [illegible] from the total pressure of 45lbs is lost, [illegible] been expanded in the cylinder to its full [illegible] the case in practice. If not expanded, the [illegible] be in proportion to the pressure of the [illegible] the atmosphere at the termination of the [illegible] pressure in the case just supposed, and [illegible] upon the piston, and expand the steam to [illegible] power of the Engine will not be doubled, [illegible] 1·161 to 1 ; because the pressure of the [illegible] makes 75, not twice 45, which would be 90. [illegible] formed, as in the Condensing Engine, steam [illegible] above the atmospheric pressure would press [illegible] 42lbs per square inch, supposing the resistance

of the uncondensed steam to be only equal to 3lbs to the square inch. As before observed, the one side of a piston of a High-pressure Engine is open to the atmosphere through the exhaust pipe, when the steam is exerting its force on the other side; and as the resistance of the atmosphere is 15lbs to the square inch, it follows that the power of the steam is as its own pressure above the atmosphere: or in other words, the pressure above the atmosphere is the available power obtained in the High-pressure or Non-condensing Engine, when the full pressure in the boiler is brought upon the piston.

The steam upon the piston of some Condensing Engines, is, at one portion of its stroke, below the atmospheric pressure. The pressure of the steam upon the piston at the commencement of the stroke or descent of the piston, is for one-seventh the length of the cylinder, equal to 32lbs to the square inch, or 17lbs to the square inch above the pressure of the atmosphere. The valve is then closed, and the communication with the boiler cut off—and the steam in the cylinder is expanded for the remainder of the stroke, decreasing as the piston approaches nearer and nearer to the end of the cylinder. Suppose the grease tap in the cylinder to remain open during the whole length of the stroke, the steam will rush out in proportion to its own pressure above the atmosphere, until the pressure is equalised. As the piston descends, the pressure of steam is reduced, by expansion, until it gets below the pressure of the atmosphere; and then the atmosphere will rush into the cylinder. At the same time the cylinder will be full of steam. After crossing the atmospheric line, the steam is not equal to the resistance, or the power of the surrounding medium; therefore, at one portion of the descent of the piston the steam will rush out, displacing the air for a certain distance of the stroke from the top of the cylinder; and at the other portion of the stroke, or of the piston's descent, the air will rush into the cylinder, when the steam is below the pressure of the atmosphere, and will exert a force upon the piston equal to the difference in pressure between the steam and the air, because the pressure of the steam in the cylinder has been reduced by expansion below the pressure of the atmosphere; but when steam below the pressure of the atmosphere is admitted into the cylinder at the commencement of the stroke, the air will rush into the cylinder at the time the steam from the boiler is admitted, if the grease-tap ¹ kept open from the commencement of the stroke to the end; t

L

[illegible] Engine [illegible] be working with steam at a less pressure [illegible] open vessel: or in other words, steam below the [illegible] atmosphere. Were the openings sufficiently large to [illegible] quantity of air, no steam could enter the cylinder [illegible] superior pressure of the atmosphere. If the steam in the [illegible] below the pressure of the atmosphere, the boiler would [illegible] with air.

[illegible] a pressure above the atmosphere is admitted into [illegible] and kept up to the end of the stroke, the steam would [illegible] grease-cap, were it open the whole length of the [illegible] the steam in the cylinder has not been reduced by [illegible] to be condensed in the condenser by an extra [illegible] water: thus giving the air-pump more work to do, and [illegible] of steam in proportion to the pressure. If the [illegible] reduced to the lowest practical working pressure by expan- [illegible] been made of it; but if not expanded to its full [illegible] be in proportion to the amount of pressure left in [illegible] previous to condensation—or the exhaustion into the [illegible] a High-pressure Engine.

[illegible] Engine not heavily loaded, working with steam below [illegible] the atmosphere from the boiler and in the cylinder, [illegible] valve lifted from its seat, air would rush into the [illegible] pressure became equal to the pressure of the [illegible] air and steam would rush into the cylinder of [illegible] force equal to the pressure of the atmosphere. [illegible] condensed, like steam, the vacuum would be [illegible] would consequently be an equal pressure on [illegible] in such case, would be a stoppage of the [illegible]

[illegible] OF STEAM.

[illegible] place when the communication with the [illegible] the cylinder of the Engine, after the piston has [illegible] stroke. The steam thus enclosed betwixt [illegible] of the cylinder, forces the piston [illegible] the stroke. Supposing the valve to have [illegible] had travelled one-fourth the length of the [illegible] at that time to be 20lbs above atmosphere

on each square inch, or 35lbs the total pressure per square inch upon the piston, it would begin to decrease in pressure as the piston were forced forward, till the piston reached the end of the stroke, when the pressure would be 8¾lbs per square inch. For one-fourth the length of the cylinder, the pressure would be 35lbs per square inch ; that is while the communication between the boiler and the face of the piston remained open. This communication being closed at that point, the steam would begin to expand, forcing the piston forward. When the latter arrived at half stroke, or at the half length of the cylinder, the pressure would be 17½lbs upon the square inch ; when it arrived at three-fourths the length of the cylinder, the pressure upon it would be 11⅔lbs per square inch ; and at the end of the stroke the pressure would be 8¾lbs per square inch, or 6¼lbs per square inch below the pressure of the atmosphere—the steam having gradually expanded during the traverse of the piston, down to 8¾lbs at the termination of the stroke—leaving only the latter amount to be condensed.

Expansion is, perhaps, the most extraordinary property of steam. The merit of the discovery is due to HORNBLOWER, who, in 1781, obtained a patent for the invention. He states that when steam is confined on one side of a piston, and a partial vacuum is formed on the other, the steam will move the piston till its force is in equilibrium with the friction and uncondensed steam on the under side of the piston, and power is communicated during the motion, in addition to the ordinary effect of the original steam pressure. To apply this power, which was *lost* before, HORNBLOWER used two cylinders in which the steam was to act. He applied the steam, after it had acted in the first cylinder, to operate a second time in the second and larger cylinder, by permitting it to expand. This he accomplished by connecting the cylinders together by proper apertures. We give his own description in the words of his specification:—"I employ the steam after it has acted in the first vessel to operate a second time in the other, by permitting it to expand itself, which I do by connecting the two vessels together."

The expansive property of steam is strictly mechanical, and is a property common to all gaseous fluids—air, gas, &c. It simply consists in this—that vapour of a given elastic force will expand to certain limits, and during the process of expansion will act on opposing bodies *with a force gradually* decreasing, causing a diminution of

elastic power in an inverse ratio to the increase of volume, until it has reached the limits of its power, or is counterbalanced by the resistance of a surrounding medium. Thus, steam of any given pressure, expanded to twice its original bulk, will exert only one half its original power. If a partial vacuum be formed on one side of a piston, its motion will be continued until the density of the steam on the other side be as low as that of the uncondensed vapour on the vacuum side of the piston. It is clear that the power which may be obtained by thus impelling a piston will be the average between the highest and the lowest pressure upon the piston. It must also be understood that IT IS A SAVING, AND NOT A GAIN, that thus results from expansion : a power being made available which was before lost, by using the steam up to its last impelling force, and not allowing it to escape until the whole of that available force has been expended. This accounts for some Engines using more fuel and steam than others, because the steam is not expanded to its utmost limit, in consequence of the steam not being cut off by the valve soon enough ; or that the load on the Engine is great, and requires the steam to be longer on the piston before it is cut off. If the load on the Engine be such as to allow the steam to be cut off early, and to expand to its full available limits in the cylinder, then the most will have been made of it ; the highest pressure in the boiler will have been used upon the piston, and down to the lowest point. Were atmospheric air compressed so as to exert a force of 20lbs on the square inch, and were the supply to be continued throughout the stroke, an impulse would be given to the piston equal to 20lbs to the square inch during the whole stroke ; but if the air was allowed to expand, the impulse would only be as the average. It will be evident that, if in the former case the air was suffered to depart from the cylinder at the same elasticity as that which it entered, we should lose the force which was necessary to compress it to its density ; while, by expanding it to its limits, we apply every part of that force. The main spring of a watch actuates its machinery in this manner:—an increasing effort is required to wind up the spring, and a decreasing impulse is given back to the machinery. But if, after the spring had partially uncoiled itself, it were then liberated, the force which wound it up to its last impelling point would be so far lost. So in the Steam Engine : if the steam be allowed to escape from the cylinder before

its force is expended by expanding to the lowest available pressure, the loss will be in proportion to the amount of the pressure not made available.

A certain quantity of fuel is required to raise steam to a certain elasticity. If that steam be allowed, after having moved the piston, to escape into the atmosphere or condenser without having acted expansively, a portion of the fuel which was consumed to raise the steam up to that point of elasticity will have been lost. In one case a given bulk of fuel would produce fifty: in the other case it would produce fifty, added to all the intermediates down to the lowest expansive force.

By this it will be apparent that the advantages arising from expansion increase with the pressure of the steam. In Condensing Engines the steam expands to half an atmosphere, and sometimes even to 10lbs or 12lbs below the atmospheric line. In Non-condensing Engines the steam expands to one or two pounds above atmosphere, but often not so low. This entails a loss in proportion to the pressure of the steam thrown into the atmosphere.

WORKING STEAM EXPANSIVELY IN ONE CYLINDER.

THERE are two modes of applying the power of steam to the working cylinder, namely: one, allowing steam to flow from the boiler during the whole length of the stroke; and the other, cutting it off from the boiler when the piston has travelled a determined distance—the great and paramount object of this last arrangement being a saving of fuel.

If steam be applied the full length of the stroke, the average pressure will be as the pressure per square inch upon the piston; but if the steam be cut off at half stroke—suppose the pressure to be 75lbs per inch when the pressure of the atmosphere is added—there will be a mean equivalent, or average pressure, throughout the stroke of 63·48lbs per square inch; being only 11·5lbs less than the full pressure, or 15·36 per cent. of a loss in power, though half the former quantity of steam has only been used. This alone effects a saving of 41 per cent., and shows the great benefit to be derived from expansion in one cylinder.

If this principle be true—and its truth is undeniable—it is quite

evident that the greatest economy will result from extending expansion to its full limit, and making the cylinders of Steam Engines of sufficient capacity for this purpose; though with the high-pressures with which expansion is most available, they will require to be less than are usually made, to allow the Engines to produce the maximum effect. The table which follows this section shows, according to the pressure used, the average pressure of steam upon the piston when cut off at at any portion of the stroke. The table begins at 10lbs above atmosphere, and advances in 5lbs up to 150lbs per square inch. This table will enable the Engineer to determine, by any given pressure, the amount of expansion required for the power to be obtained, and the saving thereby to be effected.

The principle of expanding the steam in the Condensing Engine is the same as in the Non-condensing Engine, excepting that the steam, which exhausts into the atmosphere, cannot expand below 15lbs per square inch, because the exhaust is open to the pressure of the atmosphere in all cases. The resistance of the atmosphere (15lbs) must be added to the pressure of steam above atmospheric pressure, when calculating the pressure of the expansion of steam upon the piston. For example: steam at 20lbs pressure above the atmosphere upon the piston, cut off at one-fourth the stroke, will be 8¾lbs at the termination of the stroke, as shown by the following calculation: 20lbs added to 15lbs, the pressure of the atmosphere, equals 35lbs. This divided by four, gives the quotient 8¾lbs. Thus, 8¾lbs is the pressure at the termination of the stroke, or 6¼lbs below atmospheric pressure. The diagram which follows,—No. 12, and the explanation,—illustrate the principle more fully.

The cylinder is represented by Diagram No. 12, divided into ten equal parts. The pressure of the steam used is supposed to be 60lbs per square inch above the pressure of the atmosphere. Suppose the steam to be cut off at one-tenth of the stroke, and the piston to be travelling the other nine portions by the force of the expanding steam: in calculating what will be the pressure at the termination, or at any other portion of the stroke, add the pressure of the atmosphere to the pressure of steam above atmospheric pressure. Thus, let 60lbs be the pressure of steam in the cylinder above the atmosphere, and add 15lbs as representing the pressure of the *atmosphere*, or a perfect vacuum, which together will equal 75lbs *pressure*. *Then*, whatever be the point, or proportion of the length

of stroke at which the steam is cut off, the total initial pressure must be divided by it to obtain the terminal pressure. Thus, in

Diagram No. 12.

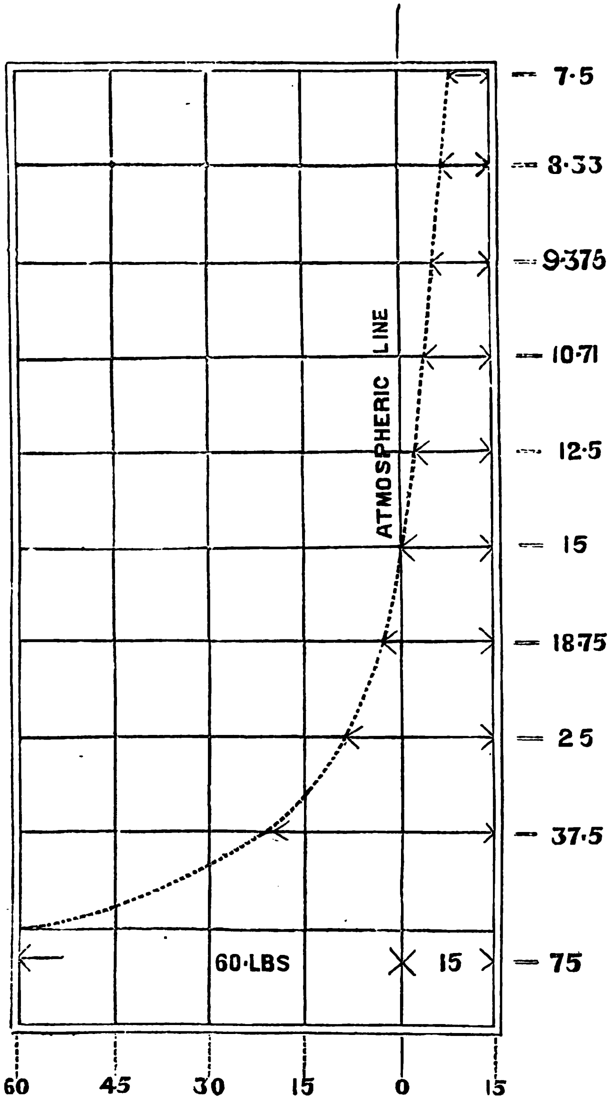


Diagram 12, it is cut off at one-tenth, and as the total initial pressure *is 75lbs, the terminal* pressure is 75÷10=7·5lbs. The

pressure is here shown at the end of each space of equal lengths through which the piston has moved. The sum of these pressures when added together, and divided by ten, will not give the average pressure, because they represent the pressures at the termination of each of the spaces, and not the average pressure of the spaces, which must be obtained to arrive at the mean average pressure of the Diagram.

The average pressure in this case—the cut-off being one-tenth, and the terminal pressure 7·5lbs, calculated by the hyberbolic logarithm, is 24·77lbs. By keeping the steam at full pressure the whole length of the stroke, the average pressure will be as the initial pressure, 75lbs. By cutting of the steam at one-tenth of the stroke, the average pressure is 24·77lbs, which is 33 per cent., whilst the quantity of steam used has been only ten per cent.,—showing the great benefit of expansive working, being in this case 33 to 10, or 3·3 to 1, in favour of expansion in this degree.

Point of cut-off.	Pressure in lbs to point of cut-off.	Terminal pressure.	Average pressure throughout the stroke.	Per centage of steam used.	Per centage of power obtained.	Area of cylinder for a given power.	Per centage of steam for a given power.
Full length.	100	100	100	100	100	100	100
$\frac{9}{10}$	100	90	99·0	90	99·0	101·01	90·909
$\frac{8}{10}$	100	80	97·84	80	97·84	102·207	81·766
$\frac{7}{10}$	100	70	95·20	70	95·20	105·042	73·528
$\frac{6}{10}$	100	60	90·0	60	90·0	111·11	66·666
$\frac{1}{2}$	100	50	84·65	50	84·65	118·133	59·066
$\frac{4}{10}$	100	40	76·64	40	76·64	130·480	52·192
$\frac{3}{10}$	100	30	65·88	30	65·88	151·791	45·537
$\frac{2}{10}$	100	20	52·18	20	52·18	191·644	38·328
$\frac{1}{10}$	100	10	33·03	10	33·03	302·752	30·275

The accompanying table shows at one view the benefit derived from the expansive properties of steam. There are eight columns and ten lines. The first column represents the steam admitted during the full length of the stroke and cut off at the end of every division. The succeeding columns give the initial pressure of the steam to the *point of cut-off,* which is 100lbs per inch, including vacuum; next,

the terminal pressure, then the average pressure; next is shown the per centage used, and then the per centage of power obtained. We have then the area of cylinder required for a given amount of power,—the area for steam at full pressure the whole length of the stroke, being 100 inches. The last column gives the per centage of steam used for a given amount of power at each point of cut-off, and with cylinders proportioned as in preceding column. It will be observed that the greater the amount of expansion, the higher is the per centage of power for a given amount of steam.

The conducting properties of the metal rob the steam of its heat in proportion to the difference of the temperature. Hence the necessity of clothing high-pressure cylinders with felt, or other non-conducting substance, to prevent the absorption of the caloric; or of casing them, keeping steam in the casings at the pressure of the boiler. The higher the temperature at which the cylinders of Steam Engines can be maintained, the better.

THE FOLLOWING TABLE

SHOWS, according to the pressure used, the average pressure of steam upon the piston, cut off at any portion of the stroke, beginning at 10lbs, and advancing in 5lbs up to 150lbs per square inch; enabling the Engineer to determine, at any given pressure, the amount of expansion requisite for the full power to be obtained, and the saving thereby to be effected. In all cases the pressure of the atmosphere must be added to the pressure of the steam above atmosphere, when reference is made to the table for the average throughout the stroke.

Example: 15lbs pressure on the piston above atmosphere, cut off at one-fourth the piston's traverse, will be thus: 15lbs steam and 15lbs the pressure of the atmosphere=30 : then look for 30lbs at the head of the table, and down the first column for $\frac{1}{4}$; trace that $\frac{1}{4}$ under 30, and you will find the average to be 17$\frac{3}{4}$lbs throughout the stroke. For other proportions follow the same principle—that is, supposing the vacuum to be 15lbs. If there be not 15lbs of a vacuum, the amount of pressure below must be deducted from the average.

If a Non-*condensing Engine,* where the steam is expanded into

A TABLE,

Showing the average Pressure of the Steam upon the Piston throughout the Stroke, when cut off in the Cylinder from ⅓ to 11/12, commencing with 10lbs and advancing in 5lbs up to 150lbs Pressure.

Steam cut off in the Cylinder.	Pressure in lbs at the commencement of the Stroke.														
	10	15	20	25	30	35	40	45	50	55	60	65	70	75	
	Average Pressure in lbs upon the Piston.														
1/3	7	10½	14½	17½	21	24½	28	31½	35	38½	42	45½	49	52½	1
2/3	9¼	14	18¾	23½	28¼	32¾	37½	42	46¾	51½	56¼	61	65½	70¼	2
1/4	6	9	12	15	17¾	20¾	23¾	26¾	29¾	32¾	35¾	38¾	41¾	44¾	3
1/2	8½	12¾	17	21	25¼	29½	33¾	38	42¼	46½	50¾	55	59¼	63½	4
3/4	9½	14½	19¼	24	28¾	33½	38½	43¼	48¼	53	57¾	62½	67½	72¼	5
1/5	5¼	7¾	10½	13	15½	18¼	20¾	23½	26	28½	31¼	34	36½	39	6
2/5	7½	11½	15¼	19	23	26¾	30½	34½	38¼	42	46	49¾	53½	57½	7
3/5	9	13	18	22½	27	31½	36¼	40¾	45¼	49¾	54¼	58¾	63¼	67¾	8
4/5	9¾	14½	19½	23¾	29¼	34¼	39	44	49	53¾	58½	63½	68½	73¾	9
1/6	4½	7	9¼	11½	14	16¼	18½	20¾	23¼	25½	27¾	30¼	32½	34¾	10
5/6	9¾	14¾	19¾	24½	29½	34½	39½	44¼	49¼	54	59	64	69	73¾	11
1/7	4¼	6¼	8¼	10½	12½	14¾	16¾	18¾	21	23¼	25¼	27¼	29½	31½	12
2/7	6½	9½	12¾	16	19¼	22½	25¾	28¾	32	35¼	38½	41¾	45	48¼	13
3/7	7¾	11¾	15¾	19¾	23¾	27¾	31½	35½	39½	43½	47½	51½	55¼	59¼	14
4/7	8¾	13¼	17¾	22¼	26¾	31¼	35½	40	44½	49	53½	57¾	62½	66¾	15
5/7	9½	14¼	19	23¾	28½	33½	38¼	42¾	47¾	52½	57¼	62	66¾	71½	16
6/7	9¾	14¾	19¾	24¾	29½	34½	39½	44½	49½	54¼	59¼	63¾	69¼	74¼	17
1/8	3¾	5¾	7¾	9½	11½	13½	15¼	17¼	19¼	21¼	23	25	27	28¾	18
3/8	7¼	11	14¾	18½	22¼	26	29¾	33½	37	40¾	44½	48¼	52	55¾	19
5/8	9¼	13¾	18¼	22¾	27½	32	36¾	41¼	45½	50½	55¼	59¾	64¼	68¾	20
7/8	9¾	14¾	19¾	24¾	29¾	34¾	39½	44½	49½	54½	59½	64½	69¼	74¼	21
1/9	3½	5¼	7	8¾	10½	12¼	14¼	15¾	17¾	19½	21¼	23	24¾	26½	22
2/9	5½	8¼	11	13¾	16½	19¼	22¼	25	27¾	30½	33¼	36	38¾	41¾	23
4/9	8	12	16	20	24	28	32	36	40¼	44¼	48¼	52¼	56¼	60¼	24
5/9	8¾	13¼	17½	22	26¼	30¾	35¼	39½	44	48½	52¾	56½	61¾	66	25
7/9	9½	14½	19¼	24¼	29	34	38¾	43¾	48½	53½	58¼	63¼	68	72¾	26
8/9	9¾	14¾	19¾	24¾	29¾	34¾	39¾	44½	49½	54½	59½	64½	69½	74½	27
1/11	3	4½	6	7¾	9¼	10¾	12¼	13¾	15¼	16¾	18¼	20	21½	23	28
2/11	4¾	7¼	9¾	12¼	14¾	17¼	19½	22	24½	27	29½	31¾	34¼	36¾	29
3/11	6¼	9¼	12½	15½	18¾	21¾	25	28	31¼	34¼	37½	40¾	43¾	47	30
4/11	7¼	10¾	14¼	18¼	21¾	25½	29¼	32¾	36½	40¼	43¾	47½	51	54¾	31
5/11	8	12	16¼	20¼	24¼	28¼	32½	36½	40½	44½	48¾	52¾	56¾	60¾	32
6/11	8¾	13	17½	21¾	26¼	30½	35	39¼	43¾	48	52½	56¾	61¼	65½	33
7/11	9¼	13¾	18¼	23	27½	32¼	36¾	41½	46	50¾	55¼	60	64½	69¼	34
8/11	9½	14¼	19	23¾	28¾	33½	38¼	43	47¾	52¾	57½	62¼	67	71¾	35
9/11	9¾	14¾	19½	24½	29¼	34¼	39¼	44	49	54	58¾	63¾	68¾	73½	36
10/11	9¾	14¾	19¾	24¾	29¾	34¾	39¾	44¾	49¾	54¾	59¾	64½	69½	74½	37
1/12	2¾	4¼	5¾	7¼	8½	10	11½	13	14½	15¾	17¼	18¾	20¼	21¾	38
7/12	8¼	12¼	16¾	21	25¼	29½	33¾	38	42¼	46½	50¾	55	59¼	63½	39
11/12	9¾	14¾	19¾	24¾	29¾	34¾	39¾	44¾	49¾	54¾	59¾	64¾	69¾	74¾	40

A TABLE,

Showing the average Pressure of the Steam upon the Piston throughout the Stroke, when cut off in the Cylinder from $\frac{1}{3}$ to $\frac{11}{12}$, commencing with 10lbs and advancing in 5lbs up to 150lbs Pressure.

	Pressure in lbs at the commencement of the Stroke.														
	80	85	90	95	100	105	110	115	120	125	130	135	140	145	150
	Average Pressure in lbs upon the Piston.														
1	56	59½	63	66½	70	73	77½	80½	84	87½	91	94½	98	101½	105
2	75	79½	84½	89	93¾	98¼	103	107¾	112½	117	121¾	126½	131	135¾	140½
3	47¾	50¾	53¾	56¾	59¾	62¾	65½	68½	71½	74½	77½	80½	83½	86½	89½
4	67¾	72	76¼	80½	84¾	89	93¼	97¼	101½	105¾	110	114¼	118½	122¾	127
5	77¼	82	87	91¾	96½	101¼	106¼	111	115¾	120¾	125½	130¼	135¼	140	144¾
6	41¾	44¼	47	49½	52¼	54¾	57¼	60	62½	65¼	67¾	70½	73	75½	78½
7	61¼	65	69	72¾	76½	80¼	84¼	88	91¾	95¾	99½	103¼	107¼	111	115
8	72½	77	81½	86	90½	95¼	89½	104¼	108¾	113¼	117¾	122¼	126¾	131½	135
9	78¼	83	88	92¾	97¾	102¾	107½	112½	117½	122¼	127¼	132	136¾	141¾	146¾
10	37¼	39½	41¾	44¼	46½	48¾	51¼	53½	55¾	58	60½	62¾	65	67½	69¾
11	78¾	83¾	88¾	93½	98½	103½	108¼	113¼	118¼	123¼	128	133	137¾	142¾	147
12	33½	35¾	37¾	40	42	44	46¼	48¼	50½	52½	54¾	56¾	58¾	61	63
13	51½	54½	57¾	61	64¼	67½	70¾	74	77¼	80½	83½	86¾	90	93¼	96½
14	63¼	67¼	71¼	75¼	79	83	87	91	94¾	98¾	102¼	106¾	110¾	114¾	118¾
15	71¼	75¾	80	84½	89	93½	98	102½	106¾	111½	115¾	120¼	124¾	129¼	133½
16	76¼	81	85¾	90¾	95½	100¼	105	109¾	114½	119¼	124	128¾	133½	138½	143¼
17	79	84	89	93¾	98¾	103¾	108¾	113¾	118¾	123½	128½	133½	138½	143½	148¼
18	30¾	32¾	34½	36½	38½	40½	42¾	44½	46¼	48	50	52	54¼	56¼	57¾
19	59½	63	66¾	70½	74¼	78	81¾	85½	89	92¾	96½	100¼	104	107¾	111½
20	73½	78	82½	87¼	91¾	96½	101	105½	110¼	114¾	119½	124	128½	133¼	137¾
21	79¼	84¼	89¼	94¼	99	104	109	114	119	124	128¾	133¾	138¾	143¾	148¾
22	28¼	30¼	31¾	33¾	35½	37¼	39	40¾	42½	44¼	46	47¾	49½	51½	53¼
23	44½	47¼	55	57¾	55½	58¼	61	63¾	66¾	69½	72¼	75	77¾	79½	83
24	64¼	68¼	72¼	76¼	80½	84½	88½	92½	96½	100½	104½	108½	112½	116½	120½
25	70½	74¾	79¼	83¾	88	92½	97	101¼	105¾	110¼	114½	119	123¼	127¾	132¼
26	77¾	82¾	87½	92¼	97¼	102	107	111¾	116¾	121½	126½	131¼	136¼	141	145
27	79¼	84¼	89¼	94¼	99¼	104¼	109¼	114¼	119¼	124	129	134	139	144	149
28	24½	26¼	27¾	29¼	30¾	32¼	33¾	35½	37	38½	40	41½	43¼	44¾	46¼
29	39¼	41¾	44¼	46¾	49	51½	54	56½	59	61¼	63¾	66¼	68¾	71¼	73¾
30	50	53¼	56¼	59½	62½	65¾	68¾	72	75¼	78¼	81½	84½	87¾	90¾	94
31	58½	62	65¾	69½	73	76¾	80¼	84	87¾	91¼	95	98½	102¼	106	109½
32	65	69	73	77	81¼	85¼	89¼	93½	97½	101½	105½	109¾	113¾	117¾	121¾
33	70	74¼	78¾	83	87½	91¾	96¼	100¾	105	109½	113¾	118¼	122½	127	131
34	73¾	78½	83	87¾	92¼	97	101½	106¼	110¾	115½	120	124¾	129¼	133¾	138½
35	76¾	81½	86¼	91	95¾	100½	105¼	110¼	115	119¾	124½	129¼	134¼	139	143½
36	78½	83½	88¼	93¼	98¼	103	108	112¾	117¾	122¾	127½	132½	137½	142¼	147¼
37	79½	84½	89½	94½	99½	104½	109½	114½	119¼	124¼	129¼	134¼	139¼	144¼	149¼
38	23¼	24½	26	27½	29	30½	31¾	33¼	34¾	35¼	37¾	39	40½	42	43½
39	67¾	72	76¼	80½	84¾	89	93¼	97½	101½	105¾	110	114¼	118½	122¾	127
40	79½	84½	89½	94½	99½	104½	109½	114½	119½	124½	129½	134½	139½	144½	149½

the atmosphere, the case will be different: because the steam cannot expand below 15lbs to the square inch,—that being the pressure of the atmosphere.

Example: 45lbs steam above atmosphere upon the piston of a High-pressure Engine, cut off at one-fourth the length of the stroke. The average pressure throughout will be—allowing 1lb for friction and back-pressure to force out the steam in the cylinder—19¾lbs. Thus: 45lbs steam cut off at one-fourth the stroke, with 15lbs added, make 60lbs. Look for 60 on the top line of the table, and ¼ on the side. Trace that ¼ to the figures under 60, and the average will be found to be 35¾lbs. Take 16lbs from 35¾lbs, for atmospheric pressure and friction, and there remains 19¾lbs — the available average pressure on the piston.

Example: 30lbs cut off at one-third. Add 15,=45. The average in the table will be 31½: deduct 16lbs, and there remain 15½lbs, the available pressure upon the piston.

Another Example: 15lbs cut off at half-stroke. Add 15,=30. The average in the table will be 25¼. Deduct 16lbs and 9¼lbs remains the average available pressure.

In these examples the steam in the cylinder has expanded to atmospheric pressure.

In proportion to the pressure of the steam, the cut off will have to be varied, as shown by the examples, if the steam is to be epxanded to its full limit in the cylinder of a Non-condensing Engine; that is down to 15lbs, or equal to the pressure of the atmosphere.

NOMINAL AND INDICATED HORSE-POWER OF STEAM ENGINES.

THE nominal horse-power of a Steam Engine, as laid down by JAMES WATT, the inventor of the *Steam Engine Indicator,* is 33,000lbs raised one foot high per minute. Upon this calculation he defined the power exerted. His first premise was to ascertain the average pressure of steam upon the piston; and for this, as a standard, he laid down as a rule, an available pressure of 7lbs per square inch. The speed of the piston he computed at 220 feet per minute. These, in his day, were about the average pressure and speed Steam Engines *were worked at.* *Thus,* a Steam Engine with a cylinder 26 inches

diameter, with an average pressure of steam and vacuum* on the piston of 7lbs per square inch—when the friction of the Engine is deducted—travelling 220 feet per minute, will be of twenty-four horse power, or, as it is termed, a twenty-four horse Engine. It would, in fact, be no more when worked at the before-named pressure and speed; that is, it would exert no more than twenty-four horse power. This then is, nominally, a twenty-four horse Engine; and were the *Indicator* to be applied, it would show the Engine to be working not more than twenty-four available horse-power; for the "indicated" horse-power in this case would be *precisely* the same as the "nominal"—viz., 24 times 33,000lbs raised one foot high per minute.

The term "*nominal* horse-power" is used because we ascertain originally the power of an Engine by calculation on the basis above laid down; and when we say "*indicated* horse-power," it is because we ascertain the power by means of the instrument known as the *Indicator*. A horse-power, whether termed "nominal" or "indicated," is therefore nothing more or less than 33,000lbs raised to the height before described—independently of the friction of the Engine.

But let us suppose the same Engine to be worked at an increased pressure— say at 30lbs per square inch average steam and vacuum; and the speed, instead of being 220, to be increased to 350 feet per minute—(and many Engines are worked at the speed and pressure here referred to)—the power of the Engine would be increased to 168 horse-power— or to seven times the amount of the "nominal" designation. Though still "nominally" a twenty-four horse Engine, it would, in reality, be an Engine of 168 horse-power. At the speed and pressure above stated, it would indicate that amount. Thus it is by the increase of speed and pressure that Steam Engines are made to exert more power than they were originally calculated for; or at least, more power than the "nominal" designation. Steam Engines are now made much stronger for the same "nominal" power, than in the days of WATT. With all Steam Engines, as the pressure and speed are increased, so is the power increased; but this augmentation of power is not obtained without an increased quantity of steam proportionate to the increased pressure, except where steam is used expansively. It will therefore be apparent that the term "nominal"

* This term is here *used as in general parlance*: not that there is POWER in vacuum; *but make the rule understood amongst* those who use the term, the popular idea is here adopted

horse-power can only be *legitimately* used in the sale or *purchase* of an Engine. On this rule a twenty-four horse Engine means that the cylinder shall be 26 inches diameter, if a Condensing Engine; and in the same proportion for a larger or a smaller Engine; for 22 square inches of area in the piston is one "nominal" horse, if a Condensing Engine. If we multiply the diameter of the cylinder by the diameter, and divide by 28, the product will be the number of "nominal" horse-power.

EXAMPLE.

Diameter of Cylinder, 30 inches.

```
     30
     30
     ——
28 ) 900 ( 32 horse-power.
     84
     ——
      60
      56
      ——
       4
```

The purchaser of a Condensing Engine should have 22 square inches area of piston for each "nominal" horse-power, whatever the speed or pressnre. The *real* power exerted can only be ascertained by the *Indicator;* hence the term "indicated horse-power."

This is the standard mode of ascertaining the real power for Condensing Engines, as laid down and established by WATT. It is now universally used by Engineers. In a case tried at Westminster, where power had been "let by the horse," it was referred to arbitration; and the question arose "what constitutes a horse-power?" After due enquiry, it was legally established, for the first time, that 33,000lbs raised one foot high per minute is one horse-power, which in the Steam Engine can only be correctly ascertained by the use of the *Indicator.*

Non-condensing, or High-pressure Engines, are calculated for the "nominal" power by the following rule:—Eleven square inches area of piston is one "nominal" horse-power. Multiply the diameter of the cylinder by the diameter, and divide by 14, the quotient will be the number of "nominal" horse-power.

A High-pressure Engine, working with 40lbs steam above atmosphere upon the piston, cut off at one-third the length of the cylinder, and expanding the remainder, the piston travelling 220 feet per *minute, would only* exert the power for which it was nominally

calculated, independent of friction; but take the same Engine and increase the speed from 220 to 440 feet per minute—which is quite practicable in a Horizontal Engine, if the fly-wheel be not too large —the power of that Engine would then be doubled; but double the quantity of steam would have been used. The exhaust steam would be thrown away at the termination of the stroke, at 3·33lbs above the pressure of the atmosphere.

Suppose steam to be introduced to the same cylinder at 80lbs pressure above atmosphere, and cut off at one-third, as in the former case, the pressure at the termination of the stroke would be 31·66lbs, or 16·66lbs above atmosphere. The result will be as follows— 40+15=55lbs pressure cut off at one-third,=18·33lbs terminal pressure, and 38·474lb absolute average pressure. Then deduct 15lbs back pressure=23·474lbs average effective pressure on the piston. Take now 80lbs+15=95lbs initial pressure, cut of at one-third= 31·66lbs terminal pressure, and 66·454lbs absolute average pressure. Then deduct 15lbs back pressure, as in the first case, and we get 51·454lbs average effective pressure on the piston. In the latter case we have 2·19 times the average pressure, whilst the steam used has been 1·727 times the quantity, so that the saving by the higher pressure is seen to be 21·2 per cent.

High-pressure Engines work at such varied pressures and speeds, that the *real* power can only be ascertained by the *Indicator;* but the "nominal" power can be calculated by the rules explained above. In the purchase of a High-pressure Engine, the buyer should have eleven square inches area of piston for each "nominal" horse-power, whatever may be the speed or the pressure the Engine can be worked at.

CHAPTER IV.

THE STEAM ENGINE INDICATOR.

HISTORY OF THE INSTRUMENT.

THE Instrument known by the name of the Steam Engine Indicator was invented by the celebrated JAMES WATT—a man whose name is indelibly associated with the improvement of the Steam Engine. The genius of WATT, combined with the facility of practical application possessed by him in an eminent degree, enabled him to become one of the greatest promoters of civilization, by placing at the command of his species a motive power for manufacturing and industrial operations almost illimitable. Having by his improvements created, as it were, this new motive power—the Steam Engine—an instrument to enable him to measure the definite amount of the power of each Engine, and to indicate defects in construction or working, became to him an indispensable requirement. To supply that want, he devised what is at present known as the Steam Engine Indicator. For a considerable period WATT kept the knowledge of that useful instrument to himself; but having at length to send a Steam Engine abroad, and the responsibility of its erection and proper working devolving upon the firm of which WATT was a member, he furnished a mechanic, whom he sent out to superintend the erection of that Engine, with an Indicator, having previously instructed him in the art of its application ; showing him that by its aid he could set the valves of the Engine so as to produce the greatest effect from a given quantity of impulsive power. It was to this circumstance that the Engineering profession are indebted for this most valuable instrument ; for, as will readily be imagined, a knowledge of the important aid it was calculated to render to the practical mechanic having been thus imparted, it could not be long kept secret, as WATT had hitherto kept it.

Since the application to the Steam Engine Indicator of the revolving cylinder, no very considerable change has been made in the instrument. Numbers of slightly varying forms and proportions have been tried, which it is not necessary here to particularize.

It is obvious that the plan of Indicator which will give the most accurate delineation of the actual operation of the steam within the cylinder, will be, undoubtedly, the best. To accomplish this, the instrument must be sufficiently free in its action to reveal every variation in pressure from whatever cause it may arise, and also sufficiently firm to ensure steadiness in working. The Swivel Arm Steam Engine Indicator, recently patented by Messrs. J. HOPKINSON & Co., Engineers, Britannia Works, Huddersfield, meets these requirements in the highest degree yet attained. It is the most recent successful improvement in the Steam Engine Indicator, and has already reached a distinguished pre-eminence. Its perfect accuracy; its complete freedom of motion,—revealing, as it does, every change of pressure, however slight; its easy adjustability, general facility of application, and its unequalled accessibility in every respect; render it altogether the most complete, perfect, and reliable Indicator ever made.

It is a fundamental principle in Mechanics and Mechanism, that in the transmission of motion a simple and direct connection is invariably the best; and any motion which is transmitted through a complex arrangement, consisting of many parts, will be obtained at considerable disadvantage. This principle applies with peculiar force to the Steam Engine Indicator, the accuracy of which is affected by the conditions here indicated. The Swivel Arm Indicator has the motion of its piston transmitted to the pencil in the most direct way possible, and without the intervention of any intermediate levers, joints, or frictional parts of any kind. All its reciprocating parts are extremely light; and, when considered relatively to the area of its piston, are infinitesimal in weight; the parts in contact are so few and simple, that the smallest possible amount of resistance is interposed to the free motion of the piston; it is, therefore, not liable to the errors and inaccuracies which more complicated arrangements invariably occasion. A multiplicity of parts for the transmission of the motion of the piston inevitably generates an amount of friction which tends to retard its action in a sensible degree, and leads the Engineer somewhat astray, who may trust to the delineations of such an instrument.

With many joints, wearing will unavoidably take place in the course of time, and this will be another source of error which it is important not to *overlook. These* considerations will show most

N

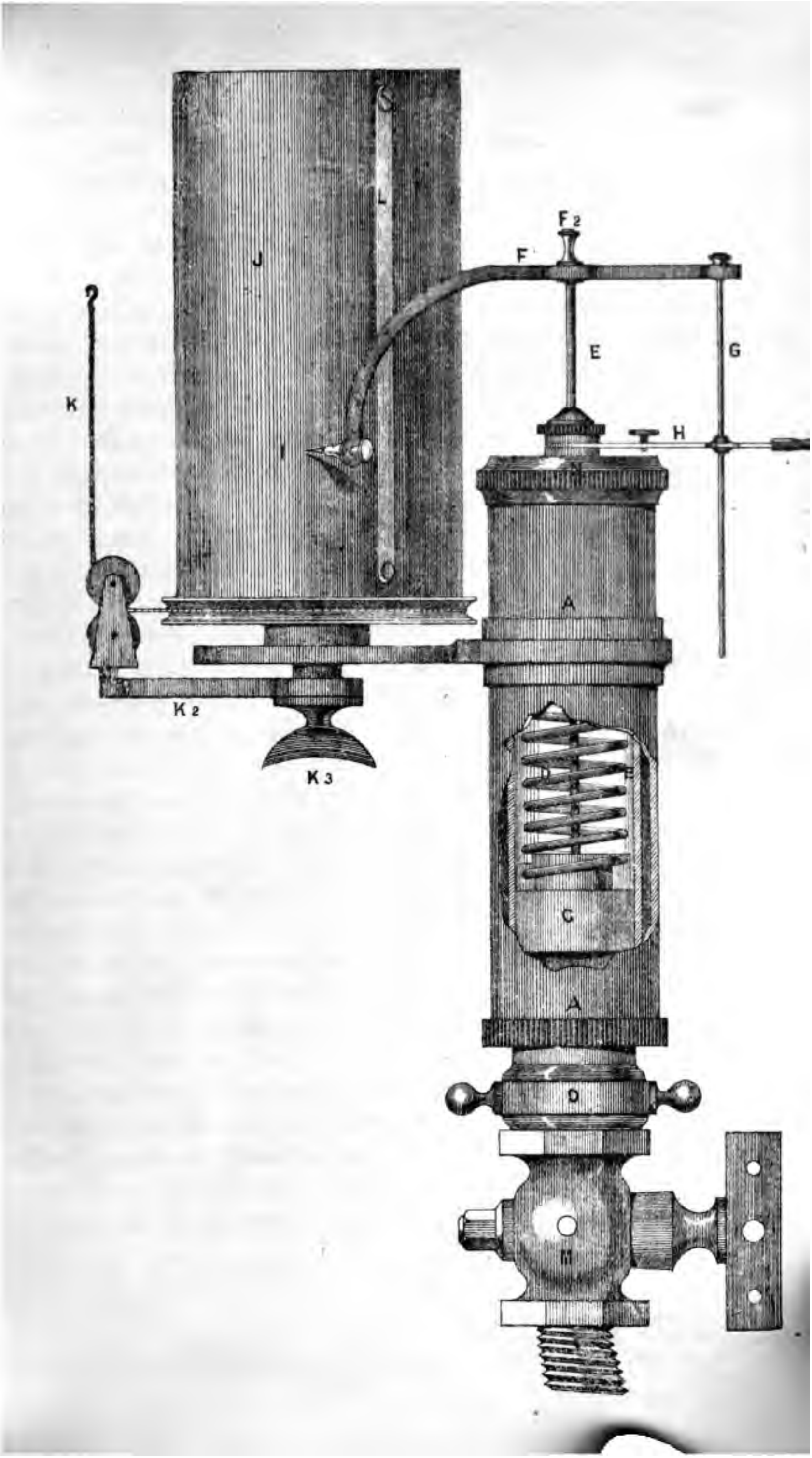
conclusively the superiority of the Direct Acting Swivel Arm Steam Engine Indicator.

DESCRIPTION OF INDICATOR.

A GENERAL view of this Indicator is given in the accompanying engraving. A is the outer barrel, or casing, of the Indicator cylinder B, leaving a space between this and the outer casing, thereby keeping the steam cylinder at a more equable temperature than would otherwise be obtained. C is the piston, working in the cylinder B, and ground to a perfect fit. D is the spiral spring, which is fast at one end to the piston C, and at the other end to a small block and disc, held firmly in position by the cap N, which is screwed on the top of the casing A. The piston-rod E passes freely through the centre of the cap N, which forms its guide, and keeps it parallel with the steam cylinder B. The swivel arm F turns freely round on the nut F^2, which screws on the top of the piston-rod E. There is a guide-bar G fixed to one end of the swivel-arm F, which slides freely through a hole (the hole being slightly elongated) in the horizontal arm H. The horizontal arm H is attached to the cap N, and turns on it as its centre, the other end of the horizontal arm H having a small handle by which the operator is enabled to apply the pencil I to the paper, which is fastened around the paper barrel J, whilst the piston-rod and swivel arm are moving up and down by the action of the steam on the piston C. The paper barrel J fits accurately on an inner barrel, which contains a helical spring, so that when the cord K has been drawn out to its proper length by its attachment to some suitable reciprocating part of the Engine, the paper barrel J will then be revolved in the opposite direction by the helical spring already named. The guide pulleys are attached to an adjustable bracket K^2, which is firmly held in whatever position may be found requisite by the thumb-screw K^3. This being adjustable, will be found to be advantageous.

The paper on which the diagram is to be taken must be secured to the paper barrel J by the bar L, which is attached by the screw at L^2; the other end being movable, and when put in the position shown, holds the paper fast.

The most convenient way of putting on the paper is to lift off



J
L
F
F 2
E
G
K
H
I
A
K 2
K 3
C
A
D
M

the paper barrel J, attach the paper, and then put the barrel in position again. M is the Indicator tap, which must be screwed to the permanent cylinder tap. The Indicator A is united to the tap M by the union coupling O, which has two short arms for the purpose of turning it round.

The method of applying the pencil to the paper is quite novel, and is one of the distinguishing features of this Indicator. It enables the operator to take the diagrams with the utmost ease and facility, and reduces the friction to a minimum; indeed, the delicacy of the guide bar G prevents any excessive friction. The Indicator is adjustable in every respect necessary for its use; the springs can be changed easily and quickly, and without removing the Indicator from the cylinder; and every part is accessible with the least possible inconvenience.*

This Indicator will be found to give the most accurate and faithful diagrams which can be obtained from a Steam Engine cylinder. The bare enumeration of a small proportion of important Engineering and Manufacturing Firms, who have already adopted it in preference to all others, would alone suffice to show its superior merits, and the high estimation in which it is already held by those who are the most competent judges. In our own and adjoining counties it is now so well known, and so highly appreciated, that no special reference is at all necessary. For more distant parts of the kingdom it may not be altogether superfluous, and certainly not inappropriate, to quote the testimony of a gentleman who is acknowledged by all Engineers, who know him, to be pre-eminently qualified to give an opinion on this subject, viz., ORME HAMERTON, Esq., Superintendent of the Board of Trade, Southampton. He says—"The little defects detected by your instrument are not observable in diagrams taken by any other Indicator."

The diagrams given in another part of this work will conclusively prove the capabilities of the Indicator, being from Engines of different speeds and pressures, and with many varieties of valves. Its extreme sensitiveness reveals every irregularity or defect of any kind; and its steadiness is all that is necessary for the most exact

* A number of springs are supplied with each Indicator, suitable for the various pressures at which steam is usually admitted on the piston. The springs supplied being respectively of 20lbs, 30lbs, 40lbs and 60lbs pressure, and=10lbs, 15lbs, 20lbs and 30lbs per inch. For example, the spring marked 20lbs is 10lbs per inch, and is suitable for all pressures in the cylinder which do not exceed 20lbs above atmosphere. So the spring marked 60lbs may be used for pressures not exceeding 60lbs above atmosphere. Springs of any required pressure are made to order.

Indication. Any increase of steadiness would necessarily be at the cost of diminished accuracy.

HOW TO USE THE INDICATOR—ARRANGEMENTS AND PRECAUTIONS NECESSARY.

To connect the Indicator to the cylinder it is necessary that there should be taps fixed permanently at each end of the cylinder for that purpose, so that the Indicator can be applied at any time conveniently. It is very objectionable to have to fix the Indicator on the grease-tap, or any such unsuitable appendage, as much loss of time is occasioned thereby: and such taps are very rarely found in positions where the Indicator can be properly worked. Before screwing on the Indicator, open the cylinder taps to which it is to be applied, in order to blow out any dirt which may have accumulated, otherwise the dirt would be blown into the cylinder of the Indicator. This precaution is absolutely necessary to ensure a correct diagram, for the Indicator cannot work properly if its piston be impeded by dirt of any kind. It is also important that the piston of the Indicator should be lubricated with the best of oil,—neat's-foot oil is found to be the best for the purpose. Pour about a dozen drops down the piston-rod when the cap is off, and the oil will get to the place where it is wanted. Take the tap which is sent with the Indicator, and screw it into the cylinder tap already named. Then take the Indicator and turn round the coupling screw O until the Indicator is firm, and so that it stands with the paper barrel in the position most suitable for the occasion.

Now, take off the paper barrel from the drum on which it is placed. Take the string which is wound round the groove at the bottom of the drum, and attach it to a string which is fastened to the radius-bar of the Engine. Let the string on the radius-bar be attached at a point which will give the right length of traverse of the paper barrel. (The length suitable for this Indicator being about six inches). In the case of Horizontal or Direct-acting Vertical Engines, a radius-bar, or some other suitable arrangement must be provided for the purpose. Put the paper on which the diagram is to be made around the loose paper barrel whilst holding it in the left hand, and fasten it down by the movable bar. Having got the right

traverse by a proper adjustment of the strings, detach them, so that you may put the paper barrel on the drum, in such a position that the notch will fit on the small projection. It will be found most convenient to take off the paper barrel every time a diagram has been taken,—being the simplest and handiest way of unfastening the paper and putting another one on.

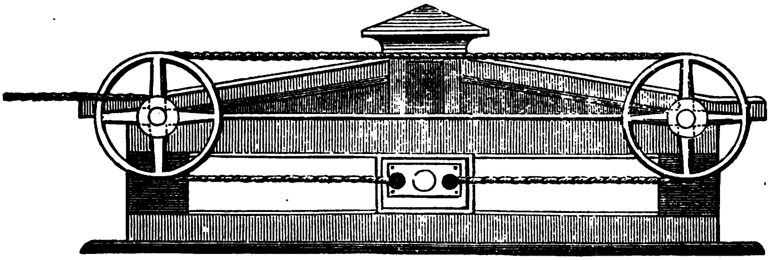
Many precautions are necessary to be observed in the attachment and working of the Indicator. In the first place, it is important that the Indicator should be fixed as close to the end of the cylinder as possible, so that the pressure on the Indicator piston will be equal to, and simultaneous with, that in the cylinder of the Engine, otherwise the diagram produced will not be correct. In some cases a pipe is brought from the end of the cylinder to a convenient place for fixing the Indicator. In other cases a pipe is brought from each end of the cylinder, meeting midway, where a two-way tap is placed, into which the Indicator tap may be screwed, so that diagrams from both ends of the cylinder may be taken without removing the Indicator. Careful experiments have proved that diagrams so taken are not correct,—showing later in opening and closing, than when the Indicator is fixed close to the end of the cylinder. It may not be altogether superfluous to say, that the cylinder taps should be so placed that when the piston is at the end of its stroke, the opening to the tap should not be covered by the piston. If this should be the case, the diagram would be useless. Numbers of cases have occurred in which taps have been so misplaced.

The next precaution to be observed is, that the spring which is attached to the radius-bar must be placed at right angles with it, otherwise the motion of the revolving paper barrel will not correspond with the motion of the piston of the Engine. Many errors arise through ignorance of this important fact; sometimes errors of a serious character. In Beam Engines—the string being attached to the radius-bar, or what is generally known as the parallel motion, will be sufficiently correct, provided the taps be fixed in suitable places,—as near perpendicular with the point at which the string is attached as is practicable. In Horizontal and Direct-acting Vertical Engines it is generally necessary to fix a small pulley for the string to pass over from the radius-bar to the Indicator, in order to obtain the proper motion. *In all cases* let this pulley be perfectly true, so

have a smooth and easy motion, so that it does not interfere with the perfect regularity of the motion of the paper barrel. See, also, that the radius-bar does not jar or vibrate, as that would communicate an unsteady motion to the rotating barrel, which must be avoided as much as possible.

The usual method of attaching the radius-bar to the cross-head is to have a slot in the radius-bar, which will slide on a stud in the cross-head,—thus allowing for the varying length from the fulcrum on which the radius-bar is centered, to the pin in the cross-head, at every part of the stroke. For Engines of quick speed it is better to attach the radius-bar to the cross-head by a connecting-rod, to allow for the variable length required, instead of having a slot as just described. This will give a steadier motion. In all cases, let the radius-bar be at right angles with the piston-rod, when the piston is in the middle of its stroke.

Diagram No. 13.



In Horizontal Engines, the radius-bar is sometimes made very short,—even less than two-thirds the length of the stroke. Now, if the Engine has an early cut-off, the diagram will not be correct, but will show the cut-off earlier on the diagram than is actually the case in the cylinder.

The deviation, or error, will be proportionate to the length of the radius-bar,—being greatest when the radius-bar is the shortest. The most correct method of giving motion to the paper barrel of the Indicator—which is applicable more particularly to Horizontal Engines,—is the arrangement of revolving pulleys, in the manner of Diagram No. 13.

Insert studs at the ends of the slides, so that the top or bottom of the pulleys be parallel with the stud in the cross-head. On these studs at the ends of the slide are to be placed grooved pulleys, suitable

for a cord of three-eighths or half inch in diameter, which must be stretched tightly over the pulleys, as shown in diagram. Fix a stud of five-eighths or three-fourths of an inch in diameter in the cross-head, having a rounded end. Then get a piece of iron about three inches long by one-and-a-half inches broad, and say three-eighths of an inch thick, and have the ends rounded as shown in diagram. Bore a hole in the middle, which will just fit on the stud in the cross-head without being too tight, but still a nice fit, so as to prevent shaking; bore a hole at each end also, into which the two ends of the cord must be fastened. The metal link thus connecting the two ends of the cord can always be put on the stud when the Engine is in motion, unless the speed should be excessive. The grooved pulley nearest the cylinder must have a boss on it, around which the string which revolves the paper barrel of the Indicator, winds and unwinds. It is quite obvious that this arrangement will give a steady and correct motion. It is much to be preferred to almost every other arrangement. The proper size of pulleys to use will depend upon circumstances. The relative diameters of the grooved pulley nearest the cylinder, and its boss, on which the Indicator string winds, can very easily be determined. Say a traverse of six inches is required for the Indicator. Suppose the stroke of the piston to be four feet. Then, six inches being one-eighth of four feet, it follows that whatever be the diameter of the pulley (that is, where the cord rests) the boss must be one-eighth of it.

It may be advisable to have a flange at the outer end of the boss, through which a small hole can be bored, in order to fasten the end of the Indicator string.

There can be no difficulty in understanding this arrangement, and applying it.

CHAPTER V.

TO FIND THE POWER OF AN ENGINE.

THE amount of power given by a Steam Engine is conventionally described as the number of horse-power. The standard of a horse-power—which is usually expressed symbolically thus, H.P., and indicated horse-power thus, I.H.P—is 33,000 foot pounds per minute; or, to state it fully, it is equivalent to 33,000lbs weight raised one foot high in one minute.

We have here three quantities—*Time, Distance,* and *Weight* To reduce the motions, pressures, &c., of the Steam Engine to these quantities, we proceed in the following way.

Ascertain the number of revolutions which the Engine runs in one minute. Then ascertain the exact length of the stroke, which can be done by measuring the distance from the centre of the fly-wheel shaft to the centre of the crank-pin; this being the radius, is half the length of the stroke. Then, as there are two strokes in one revolution, so double the length of the stroke, or four times the radius, must be multiplied by the number of revolutions to obtain the number of feet which the piston traverses in one minute.*

We have now got two of the quantities—*time* and *distance;* the remaining quantity, *weight,* we obtain thus: Get the diameter of the cylinder in inches; then square the diameter, and multiply the product by the decimals 0·7854, which will give the area of the cylinder in square inches.† In order to be absolutely correct, the sectional area of the piston-rod, which is non-effective, should be deducted from the area of the cylinder. If the piston-rod be only at one end of the cylinder, as is generally the case, then half its sectional area only should be deducted from the area of the cylinder, in order to get the mean of both ends. Having now got the correct area,

* Much confusion arises from incorrect definitions. The old practice (which ought to be obsolete now) of speaking of a revolution of the Engine as a "stroke," is entirely wrong and inapplicable to Engines which give a continuous rotary motion. Each time the piston moves from one end to the other end of the cylinder, constitutes a stroke. Engineers should remember this, and be more precise in their statements. The old definition applies only to the Single-acting Pumping Engine.

† *The area can readily be got* from the Table in the Appendix.

take diagrams from each end of the cylinder, in the way already described. Divide the diagrams into ten equal parts by the brass dividing scale, and measure each space by the pressure scale—taking care to use the scale corresponding with the spring used when the diagram was taken. Be careful to get the mean pressure in each of the spaces, and write it down in figures in the space. In diagrams from Condensing Engines, it is a general practice to measure the spaces above the atmospheric line, and the spaces below, separately, and add the sums of the two columns together. For the purpose merely of ascertaining the effective pressure, one measurement will do equally well. The measurement of the power is the area of the enclosed space within the bounding lines of the diagram made during one revolution of the Engine, so that for this purpose the atmospheric line might be disregarded.

Having now got all the spaces measured, add them together, and place the sum of the whole at the foot, and divide by ten—the number of spaces. The simplest way of dividing by ten is to place the decimal point. If the total be, say 145, then place the decimal point thus, 14·5, which is all that is required. If the pressure should be found unequal at the two ends of the cylinder, then it will be the simplest to add together the averages of each end, and divide by two, in order to get the mean pressure of both ends.

The mean average pressure of steam on the piston, as per diagrams, being now obtained, multiply it by the area of the cylinder (or piston), and this will give the number of pounds pressure on the piston. This is the third quantity—*weight.*

Having now got all the quantities, we have only to multiply together the two latter—*distance* and *weight*—and divide by 33,000, and we obtain the indicated horse-power (I.H.P.)

Diagrams will occasionally be found which seem not to be answerable to the conditions here given for their measurement. Such is No 40, which has two enclosed spaces. In this, and in all similar cases, the smaller enclosed space must be deducted from the larger one. Now, as every reader may not see why this deduction should be made, we will explain.

When the piston is advancing, the amount of the pressure propels it, in excess of the back pressure, constitutes the effective pressure ; for if the advancing pressure and the back pressure were equal, the piston would *simply be in equilibrio*—that is, it would have the

pressures of each side of it balanced, and therefore would not move, unless moved by some external power. In Diagram No. 40, the line of propulsion is seen to terminate at 18lbs above the atmospheric line; whilst the back pressure line rises to 22lbs, or 4lbs higher, before the piston begins to return. From this end of the diagram to the point of intersection of the lines, the retarding pressure is greater than the propelling pressure, and must, therefore, be deducted from the other part of the diagram to give the true effective pressure on the piston. In measuring this diagram, divide the whole length into ten equal parts, as in all others, and note down the average of each space. Then deduct the sum of the minus quantities (the sum of the spaces at the small end of the diagram) from the sum of the positive quantities (the sum of the spaces at the larger end of the diagram), and divide by ten, as in all other cases. This will be a guide to the understanding of all other diagrams in which one part must be deducted from another.

It was formerly the practice to make a deduction for the friction of the Engine—an average of 3lbs being usually allowed. The real amount will be a very variable quantity, as will be seen hereafter. This practice has now been very generally abandoned, and the total of the diagram is taken as the measurement of the power of the the Engine. In all cases where calculations have to be made repeatedly for the same Engine, an easier method of procedure may be adopted than the one given above.

The cylinder being of a given area, and the speed of the piston being a given number of feet per minute, all that is necessary is to multiply the area of the cylinder by the number of feet traversed by the piston, then divide by 33,000, and the quotient will be the I.H.P. per pound pressure. This number being always borne in mind, then, whenever diagrams are taken, and the average pressure found—the constant number has only to be multiplied by the average pressure, and the total I.H.P. will be obtained.

Let us take an example. Suppose the cylinder to be 30 inches diameter, the stroke 5 feet, and the speed 40 revolutions per minute. Now, 5 feet stroke=10 feet per revolution; and as the speed is 40, the piston will pass through 400 feet per minute. The diameter of cylinder being 30 inches, the area will be 706·86 square inches. Assuming the piston-rod to pass through both ends of the cylinder, *and to be 4½ inches* diameter, the sectional area of which is 15·9

inches, which being deducted from 706·86, given above, the effective area of the cylinder will be 691 square inches. Then $691 \times 400 = 276{,}400 \div 33{,}000 = 8·375$ I.H.P. per pound pressure. Now, if the diagram should give an average pressure of 20lbs, then we should have 8·375 I.H.P. × 20lbs = 167·5 I.H.P. Taking again the above quantities—that is, the speed, dimensions, and pressure of an Engine—and putting the two equations in the usual recognised form, they will stand thus :—

1st—$\dfrac{691 \times 20 \times 400}{33000} = 167·5$ I.H.P.

2nd—$\dfrac{691 \times 400}{33000} = 8·375$ I.H.P. $\times 20 = 167·5$ I.H.P.

The instructions above given for deduction of the sectional area of the piston-rod may seem to some Engineers superfluous, as it is rarely done; but its importance in some cases is too great to be neglected, as a very sensible difference in the power and in the economy of the Engine will sometimes be found. As an example, we know of a Compound Engine in which the high-pressure cylinder is seven feet stroke and twelve inches diameter, and the piston-rod four and a half inches diameter. Now this amounts to one-seventh of the area of the cylinder ; and, therefore, whatever might be the power of the cylinder, calculated in the usual way, without deduction for piston-rod, a deduction of one-seventh of the power must be made to obtain the true power.

It may sometimes be desirable to ascertain the I.H.P of a limited portion of the total power which an Engine has to drive. Such, for instance, as one room in a mill—a certain machine, or class of machinery—and various other circumstances which are found to obtain. All that is requisite is to take diagrams with the full load, and then with such specified portions disconnected, and by subtracting the I.H.P. of the latter from the former diagram the desired result will be obtained.

A further precaution is desirable to be observed. As the friction of the Engine is inevitably a part of the total power exerted, then a fair apportionment of it would require to be added to the amount of power shown for such portions of machinery.

CHAPTER VI.

FRICTION DIAGRAMS.

MILLOWNERS and Engineers are strongly recommended to take friction diagrams at short periodical intervals, in order that they may be able to reduce the friction of Engine and shafting to a minimum. It is highy probable that in many cases the friction might be considerably reduced. How excessive friction may arise will be easily seen by any one who has a moderate amount of experience and intelligence. Such things, for instance, as the piston-rings being screwed up too tight. Many Engineers, no doubt, remember instances where this has been done to such a degree that the Engine could not be run at the full speed, and it has been necessary to stop the Engine and slacken the piston-rings before the proper speed could be obtained. And so also in regard to the air-pump Then, it is equally important to attend to the shafting ; and, indeed, it is rarely practicable to take a friction diagram of Engine alone, but it must be taken in connection with the shafting. Now suppose that the bearings of the shafts should be disturbed, and thereby the shafting should be thrown out of the true line, it is evident that a considerable amount of friction will be generated.

The Indicator may also be made the most efficient of oil testers, by taking diagrams regularly, when different kinds of oil are being used. And assuming that all other things are equal (and great care should be exercised to ensure this) this will be a very reliable test of the lubricating properties of the oil.

It will be obvious, on a consideration of the subject, that friction diagrams will vary immensely in different Engines according to the particular circumstances of each case. If the diagram must be taken with the shafting, there will necessarily be variable conditions according to the purpose to which the Engine may be applied, and which will greatly affect the result. Now, it would be manifestly wrong to take the shafting in any of our manufacturing concerns as friction ; because it is an integral part of the weight to be driven, *and as such ought to* be estimated along with the machinery, as it

must always be included with the weight so driven, whatever may be the motive power applied; therefore, the Engine should be disconnected from the shafting to obtain proper and correct friction diagrams.

In corn mills (in many districts called flour mills) the power required to drive the shafting when the belts are off, will be small in comparison with a mill for spinning and manufacturing purposes. Even in cotton mills the friction diagrams range from five to twenty-five per cent. of the gross total power. The friction of Engines will vary considerably as the effect of many other circumstances. For instance, an Engine which is well made, and which has all its reciprocating parts perfectly true and smooth, and properly lubricated, will generate much less friction than the converse of this. A small Engine of quick speed will generate less friction than a large Engine of slow speed—assuming each to be driving the same amount of power—because the amount of friction is determined by the pressure and the nature and state of the surfaces in contact only, and not by the speed. So likewise a complex Engine—that is, one having a great number of moving parts in contact—will give a greater amount of friction than one of simple construction and few parts.

The amount of friction generated by the motion of the valves is sometimes a serious item. Take, for example, an ordinary Three-port Slide Valve, with which every Engineer is familiar. Suppose the cylinder to be thirty-six inches in diameter, and the port to be twenty-four inches long, which is a good proportion —being two-thirds of the diameter of the cylinder. Then let there be one inch at each side of the valve in addition to cover the ports properly, making it twenty-six inches wide. We will say the length is twenty inches—that is, in the direction of its motion. The whole area will than be 520 inches. Now let us suppose the pressure of steam in he valve chest to be 60℔s per inch, and an average vacuum in the cylinder of 10℔s per inch. If the valve be a perfect fit on the fixed face --that is, if the two faces are perfect planes, so that they are in close contact in every part where the surfaces are supposed to be in contact—then the pressure of the steam in the valve chest will be exerted on the whole area, minus the area overlapping the port at the steam end of the cylinder for the time being. Now the pressure beneath this port of the valve will vary much in different cases.

If the steam be kept on during the whole length of the stroke, then the valve, not covering the steam port, the pressure will be on the whole valve. If the steam be cut off very early, then the pressure in the cylinder being rapidly reduced, the pressure beneath the part of the valve covering the port at the steam end becomes a very small amount. With these considerations, it will be seen that to assume the pressure beneath the whole area of the valve at 10lbs below atmosphere, will not be overstating the case. The area of the valve being 520 inches, and the pressure of steam in the valve chest 60lbs, to which we must add 10lbs vacuum=70lbs effective pressure per inch. The total pressure on the valve will, therefore, be $520 \times 70 =$ 36,400lbs—being more than sixteen tons.

The question of friction here is a very important one, as will be evident to the commonest apprehension. Who, that has seen much of steam Engines, has not often noticed the great strain and vibration shown by the eccentric rod. And how common is the occurrence of a rocking shaft breaking—simply by torsion. These facts, without the calculations which we have given, indicate the great amount of power which is here absorbed.

For the further consideration of this subject the reader is referred to the chapter on the Construction and Action of Valves.

CHAPTER VII.

COMPUTATION OF THE ECONOMY OF STEAM ENGINES.

IN computing the economy of an Engine, it is important to exercise great care and discrimination, in order to obtain even approximately accurate results. Whoever has had much experience in this department of Engineering will have discovered the numerous sources of liability to error, and the extreme difficulty of arriving at a correct conclusion.

In comparing the relative economy of different cases, it is necessary to take into account, and estimate fairly, all the circumstances of each particular case, otherwise the apparent may not correspond with the real and true result. For the information of those who may not know, we may state that the way in which the economy of a Steam Engine is computed, is to ascertain the mean average I.H.P. for a given time—say a day, or a week, or a month—and, having ascertained the weight of coal consumed during the time, divide the total weight of the coal in pounds by the number of hours which the Engine has been running, and this will give the number of pounds of coal consumed per hour; then divide this by the number of I.H.P., and the consumption of coal per I.H.P. per hour will be obtained. The weight of coal per I.H.P. per hour is accepted as the test of the degree of efficiency of an Engine and boiler.

So far this seems simple enough. In any district where the nature of the work in different establishments is the same, a fair comparison may be made amongst them. But when we proceed to compare the economy of these with Engines in another district, engaged in another kind of trade altogether, we are generally at fault. Take the case of a cotton manufacturing concern, in which a large amount of steam is used for keeping the rooms warm, and for keeping the steam in the boiler, and maintaining a rather high temperature in the mill continuously day and night. Now all these requirements will inevitably make up a large item in the gross consumption of coal. If a separate boiler was used for these purposes, and a *record kept* of the weight of coal used for powe

only, then there would be no difficulty in correctly computing the economy of the Engine. In the case of a corn mill, the whole of the steam raised goes to the Engine for the production of power. If, in addition to this, the Engine be kept running day and night, the greatest economy can be obtained.

We frequently see reports on the economy of Ocean Steamers, in which a very high degree of efficiency appears to be attained. In comparing these with results obtained in manufacturing establishments, it should always be remembered that the Marine Engines run continuously, and that no steam is required for any other purpose than that of giving power; and, furthermore, that the best quality of coal is invariably used, whereas in the manufacturing districts it is almost as invariably the cheapest which can be obtained.

There is a very general tendency amongst Engine-drivers to over-estimate the power which their Engines may be giving. Now this is highly objectionable and reprehensible, and ought by all means to be scrupulously avoided. Every one knows that the load on the Engine will, in almost all cases, be variable. If the calculations of power be made from the diagrams which have been taken from the Engine under its greatest load only, than the computed economy will be incorrect. In all cases endeavour to obtain the mean power for any given period or lapse of time.

So far we have been considering the question of the computation of economy of Engine and boiler in the aggregate. But it is desirable to know the efficiency of each of these components. Unless both of them be efficient, it is clear that the most economical results cannot be attained. It is obvious, therefore, that in order to arrive at the greatest efficiency and economy, it is absolutely necessary that both Engine and boiler be of good construction, and in good condition. If one boiler evaporates only 6lbs of water per pound of coal, whilst another boiler evaporates 12lbs, then, with Engines exactly similar, there must be a difference of one-half in the real economy of the boilers, and in the apparent, though not in the real, economy of the Engines.

This fact should always be remembered in computing the economy of an Engine. It will be seen that in all cases it is desirable to know the evaporative duty of the boiler.

CHAPTER VIII.

THE CONSTRUCTION AND ACTION OF VALVES.

IT was originally intended that this chapter should contain descriptions and illustrations of all the best, as well as all the best known forms of Valves. This project, however, has been abandoned for many reasons. In the first place, a volume would be required to do ample justice to the subject; and in the second place, without such extensive treatment, the selection of a limited number of special forms would seem like a partial and invidious distinction amongst a great number of deserving claimants. By such a course we should be unintentionally doing an injustice to many plans and parties, which we desire to avoid. We shall therefore treat the subject in a general and comprehensive way, stating, as clearly and as briefly as practicable, the principles which should be kept in view in the construction of the Valves of Steam Engines.

Since the time that HUMPHREY POTTER first made the Valves of the Steam Engine self acting, by connecting the Valve-cock handles to the reciprocating parts of the Engine by cords, an almost illimitable variety of forms of Valves and Valve motions have been invented. The Three-port Slide Valve is the plan which is the most general; not because it is the best, but because it is the simplest and cheapest.

For the sake of the uniniated, however, we will give a few illustrations of the most common forms of Valves, which are to enable such readers to see the general principles of action, and the better to understand the various forms of Indicator diagrams.

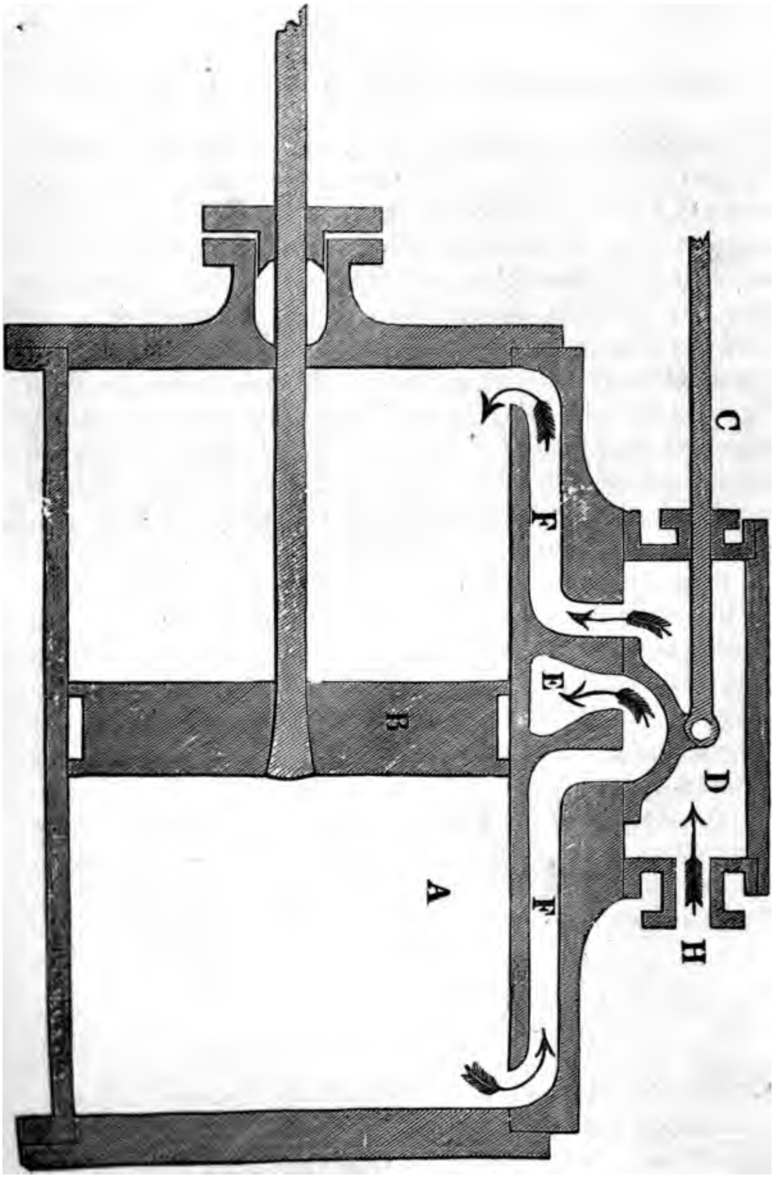
COMMON THREE-PORT SLIDE VALVE.

DIAGRAM No. 14 is an illustration of a Steam Engine cylinder with a Three-port Slide Valve, without "lap." The arrow at H denotes the admission of steam into the Valve chest, from thence to the cylinder, and from *the cylinder*—after having exerted its force upo

P

Diagram No. 14.

CYLINDER AND SLIDE VALVE.



the piston—through the Valve and exhaust port to the condenser—or into the atmosphere, if a Non-condensing Engine.

On reference to the illustration it will be seen that the steam and exhaust ports are of equal dimensions; F F are the thoroughfares to each end of the cylinder, and E the thoroughfare for the exhaust steam to escape into the condenser or into the atmosphere, as the case may be.

The Valve, as now shown, is attached to the rod C, the traverse being attained by the usual means. When the Engine makes one-half of a revolution, or one stroke up or down, the Valve changes its position accordingly.

As before observed, the arrow H denotes the steam passage from the boiler to the Valve chest D. As the Valve is shown, there is a direct opening for the steam through the steam-way F, to act upon the piston B; and on the under side A, through the exhaust way F, there is a free passage for the exhaust steam into the condenser or atmosphere, as shown at E. The Valve being hollow on the under side, permits the egress, as shown. It will be observed that the faces of the Valve will little more than cover the ports, and this construction is, therefore, appropriately termed "without cover" or "lap."

Suppose, then, that the piston had proceeded in its traverse to the end of the stroke, the Valve would also have changed in position and action; the steam from the steam chest would be admitted at the other end of the Valve on to the other side of the piston, and the steam, which in the former position of the Valve had forced the piston to the end of the cylinder, would now be liberated, and allowed to exhaust or depart into the condenser, or atmosphere, through E. The traversing motion of the Valve, which is effected by the eccentric, thus changes the application of the steam to each end of the cylinder, provides an inlet for the steam to the piston, and also an opening for exhaustion through the proper channels, after power has been exerted, producing a continuous backward and forward action of the piston of the Engine, and, by means of a beam and connecting-rod attached to the piston, communicating motion to the crank, or in some cases direct to the crank by a connecting-rod only. In the latter case the Engine is called a Direct-action Engine.

The regularity of speed and economy in the working of the Steam Engine depends in a great degree upon the proper arrangement of the Valves. *It is of the utmost* consequence that full and comple

information on this branch of the subject should be acquired by all who are entrusted with the care and direction of this valuable instrument of power. Many Engines work inefficiently, not only from the defective construction of the Valves, but also from imperfect "setting." It will work inefficiently if the thoroughfares are contracted, or the Valve-box be too short, or both. In such cases there is little chance of effecting much improvement in the working.

The Three-port Valve described above is, as we have said, without "cover" or "lap," being only just sufficient to cover the ports, and prevent the steam passing into both ports at one and the same time. It will be observed that as one port closes the other opens, and the exhaust also. Valves constructed and set in this manner admit steam upon the piston the whole length of the stroke; the exhaust is also open for the full length of the stroke.

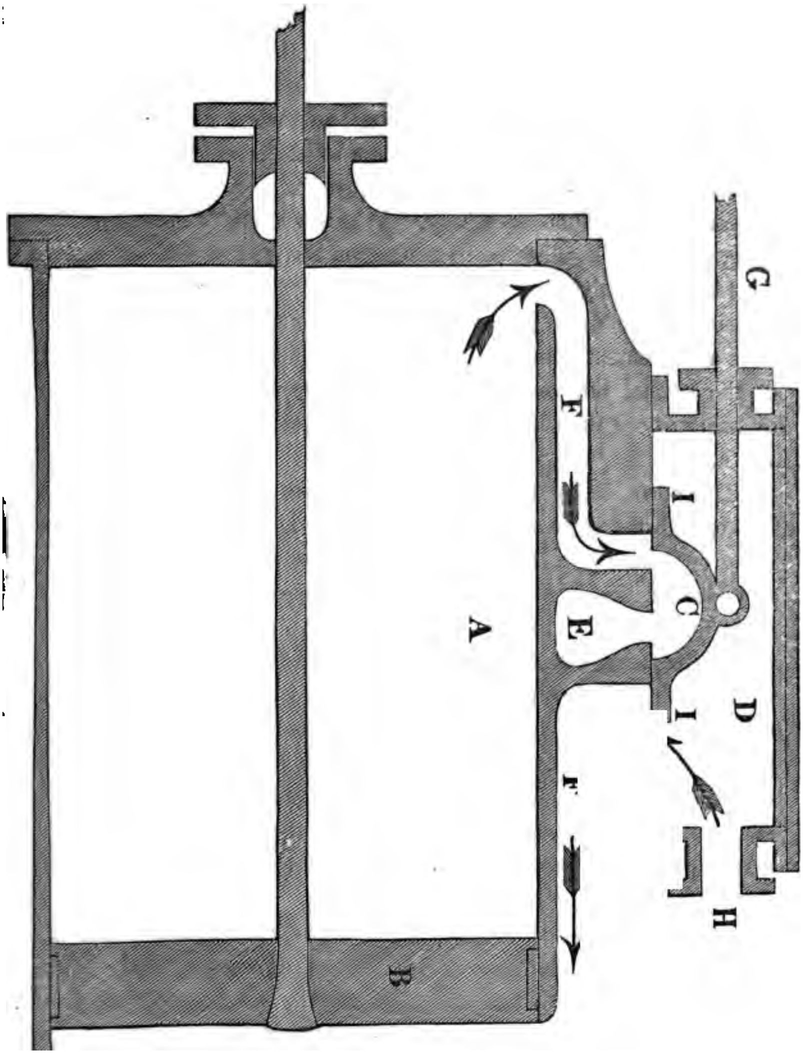
To enable this Valve to cut-off earlier and work expansively, without adding "lap," would be to make up part of the thoroughfares at each end of the Valve-box towards the outer edge, and cut away some of the metal in the inside of the ports, so that the ports would not be much contracted. In that case the Valve would cut off the steam the same as if it had "lap." The hollow of the Valve would require to be proportionately less, by being "bitted" in the same way as the thoroughfares. This mode is resorted to when the Valve-box is too short, and where "lap" cannot be added to the Valve; this is called "lapping the thoroughfares."

"LAP," OR COVER OF THREE-PORT SLIDE VALVES.

DIAGRAM No. 15 is a representation of a cylinder and a Three-port Slide Valve. The same letters denote the same parts as in Diagram No. 14. In this instance the Valve is shown with "lap" or cover, so as to cut off the steam to work expansively. This latter is accomplished to a greater or less degree according to the amount of "lap" or cover of the Valve. It will be observed in Diagram No. 15 that the cover is three-fourths broader than the steam ports, and being worked by the same eccentric, with the same length of traverse, it does not open the steam port more than one-fourth of its width, which in some cases would not be sufficient to admit the full pressure of steam to the piston; but where the ports are of sufficient width,

Diagram No. 15.

CYLINDER AND SLIDE VALVE, WITH "LAP."



and there is a proper pressure in the boiler, no difficulty is experienced by adding "lap" to the Valve.

Let us ascertain the effect of this additional breadth of Valve face. When the Valve edge arrives at the port, ready to admit the steam upon the piston, it will have moved three-fourths the length of its traverse. The remaining distance is the amount of opening given for the admission of steam to the piston, which will not be more than half an inch, if the ports be two inches wide: the Valve is then reversed by the motion or throw of the eccentric, and it will therefore have opened and closed the steam port with one inch of traverse, half an inch each way. As this Valve is V shaped on the edge, the steam is admitted to the piston easily, and cut off easily. This is technically termed "easy steaming." The full length of the traverse of the Valve is four inches for one stroke of the Engine. This arrangement cuts off the steam at about one-fourth the length of the stroke, the remainder of the stroke being worked by the expansibility of the steam. Each end of the Valve is of the same width.

The ports, being two inches wide, require that the face of the Valve, with the same traverse, shall be three and a half inches broad, being one and a half inches broader than the width of the ports. It will also be perceived that the exhaust side of the Valve will be three-fourths open when the crank is at the plumb centre. By these means the steam in the cylinder is exhausted ready for the return stroke. The closing of the exhaust is also effected much earlier, whereby the uncondensed steam left in the cylinder becomes compressed towards the end of the stroke. This in practice is found to act as a buffer to the piston, preventing a sudden shock at the termination and commencement of the stroke. The steam thus compressed is not lost, but gives back a portion of its power on the reversal of the piston.

It is necessary for an Engine running quickly, with rapid reversals of the piston, that the Valve should cut off the steam early in the cylinder, and the exhaust also close early. The example we have given is considered an extreme amount of "lap" on a Three-port Valve. Other proportions of "lap," less or more, follow the same principle: that is, if the "lap" or cover be less, the steam will be longer on the piston before it is cut off. The opening and closing *of the exhaust will* be in like ratio. The traverse of the Valve may

be increased by a new eccentric, having more throw, or by moving the rock-shaft stud nearer to the centre, providing the Valve-box be of sufficient length to enable the Valve to travel a longer distance in the same time, by which the ports will be opened wider for the admission of steam to the piston than with its former traverse; and the steam will be cut off at the same point as before, because the Valve travels quicker. This is usually termed "travelling over port."

This form and description of the Three-port Valve and Slide is principally used for High-pressure Engines. Various may be the alterations required, according to the width and distance of the ports and the length of traverse, which may be either increased or decreased, if worked by a rock-shaft. The makers of small portable Engines will find that attention to the form and setting of Three-port Valves will be attended with advantage by admitting the steam to the piston easily. They should also allow plenty of room in the thoroughfares, so that in the process of exhaustion there be no back pressure, that the speed of the Engine may be increased when more power is required.

THE D SLIDE VALVE.

THIS description of Valve, which is in very general use for Condensing Steam Engines, when first introduced was worked without "lap," or extra cover, and the Valve-boxes were generally of such a length that they would not admit of the Valve travelling over port, consequently, the steam was admitted the whole length of the stroke, which caused the exhaust side of the Valve to be too late both in opening and closing. For a long period Low-pressure Engines worked in the manner described; but as the property of the expansion of steam became better understood, various methods were adopted whereby the D Valve could be made to work on the expansive principle. It was found that by enlarging the Valve-box to allow of more traverse, thus giving room for extra "lap," the steam could be cut off at almost any portion of the traverse. This fact was partially known previous to 1841; but, in that year, a Mr. BOULD, cotton spinner, of Halifax, in Yorkshire, took out a patent for "Improvements in *Condensing* Steam Engines," in which he

describes the D Valve and the mode of "lapping." As it appears from his specification that he clearly understood the subject of "lapping," and setting Valves, extracts from that specification, in his own language, will be found of advantage. He says :—

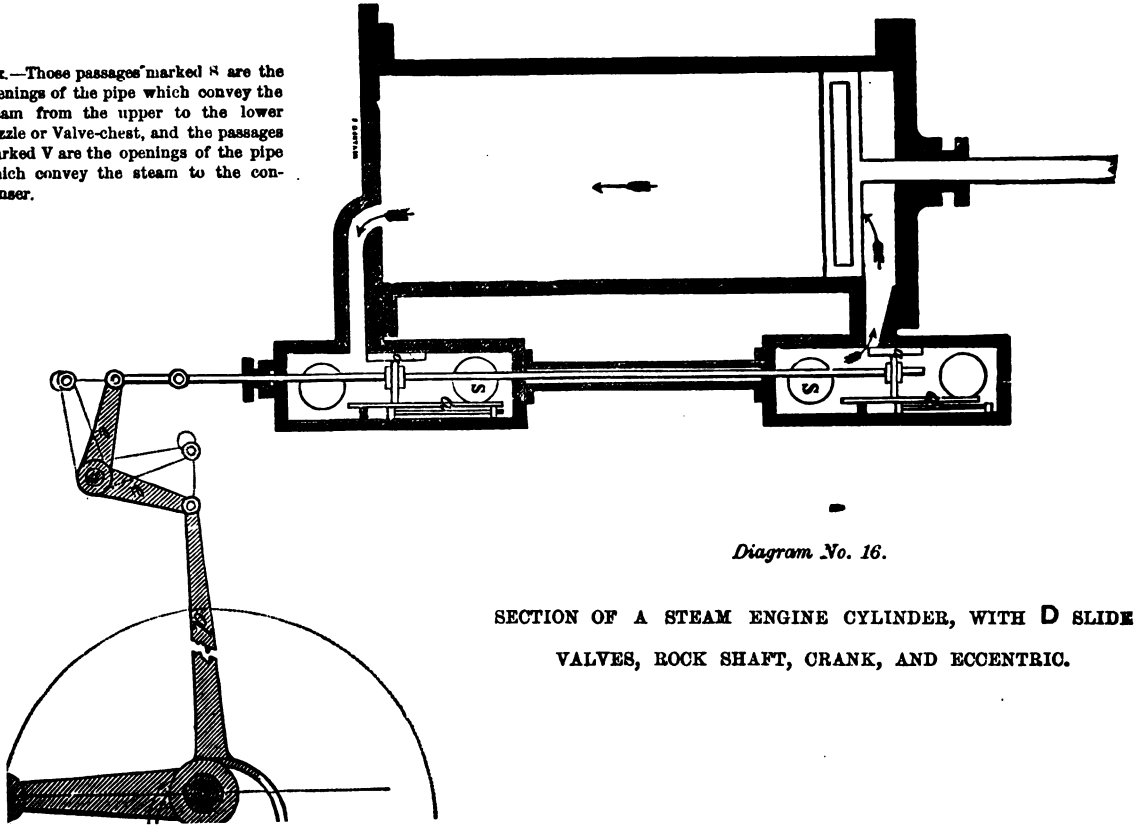
"My invention relates to the mode of arranging the Valves (which slide over the ports) of Condensing Steam Engines, in such a manner that the exhaustion port shall be fully open at the time the piston has completed its stroke, at either end of the cylinder, and at the same time allow of the steam being worked expansively by means of the same Valves. * * * In constructing and applying Valves to Condensing Steam Engines, the Valves of which are worked by eccentrics, it is usual to form the Valves of such dimensions that they will only just cover the ports or openings into the cylinder, or only very slightly exceed the dimensions of the port; hence, when the Valves are moved, the ports are simultaneously, or nearly simultaneously, opened and closed. * * * By enlarging the size of the slides so that they will be much larger (at least twice as large as the ports they work over), the Induction Valve may remain closed during a considerable length of movement of the Valves; hence it may be closed at any desired part of the stroke of the Engine, and the Exhaustion Valve remain open to the condenser; and what is also very important, the Exhaustion Valve may be fully open at the termination of the stroke at either end of the cylinder, so that a good vacuum may at the commencement of the stroke be obtained, and consequently at the time the induction port is open for the flow of steam.

"Diagram No. 16 represents a steam cylinder and Valves, and also the eccentrics and parts which work the Valves. In Diagram No. 16 the Valves are shown in the position they would be at the termination of the stroke at the upper end of the cylinder; and it will be seen that the Exhaustion Valve has passed away from the exhaustion port, and the Induction or Steam Valve has partially opened the induction port. * * * That is not necessary; but if desired, it will be evident to the Engineer that that might be done either by increasing the throw of the eccentric, or by shortening the lever E^1, to which the eccentric rod is attached.

"Diagram No. 17 shows a section of a steam cylinder and Valves, but not the lever which works them, or the eccentric which gives *motion to the lever*; the Valves in this diagram being in the

reverse position to that shown in Diagram No. 16. The Valves A A are made of such length as to be equal to three times the opening of the ports; and it will be seen by tracing the movement which will take place in the parts of Diagram No. 16, that when the stroke in the upward direction has been completed, and the Steam or Induction Valve opened to the required extent, and steam admitted above the piston, the Steam or Induction Valve will be closed by the time the piston has moved about one-third of its stroke; but this may be varied, and the exhaustion port will remain open, and the movement of the eccentric, and the parts actuated thereby, will, near the end of the stroke, close the exhaustion port at the upper end of the cylinder, and bring the parts into the positions shown in Diagram No. 17; and it will also be seen that these effects will be consequent on the enlargement of the Valves in respect to the ports over which they work, and the extent of movement given to the Valves, whereby one or other of the Valves can be moved for a considerable length whilst over its port, without uncovering that port, allowing the exhaustion port to be fully open at the time of completing the stroke at either end of the cylinder. At Diagram No. 16, B is the eccentric on the main shaft C, and the rod D and levers E and E^1 are for moving the slides. These are of the ordinary construction, allowing only for the quantity of motion given to the Valves. And it is important to call attention to the circumstance of the distance over which the Valves are moved, as on that circumstance, combined with the length of the Valves, depends the beneficial results obtained.

"Diagrams 16 and 17 show the parts so arranged as to cause the Valves to move a distance equal to four times the depth of the ports; and if it were desired that the steam port should be opened fully, then the Valve would have to move five times the length of the depth of the port; and in case of making the Valves less, to the extent of their being twice the length of the depth of the ports. Then the movement given to them is to be four times the depth of the ports to fully open the steam ports. The movement ought not to be reduced three times the depth of the ports, as the steam port will then be only half opened. It should be stated that although I prefer to work the slides with an eccentric, it will be evident that a crank may be substituted. * * * Although I have shown the Valves as being equal to three *times the length* of the depth of the ports over which



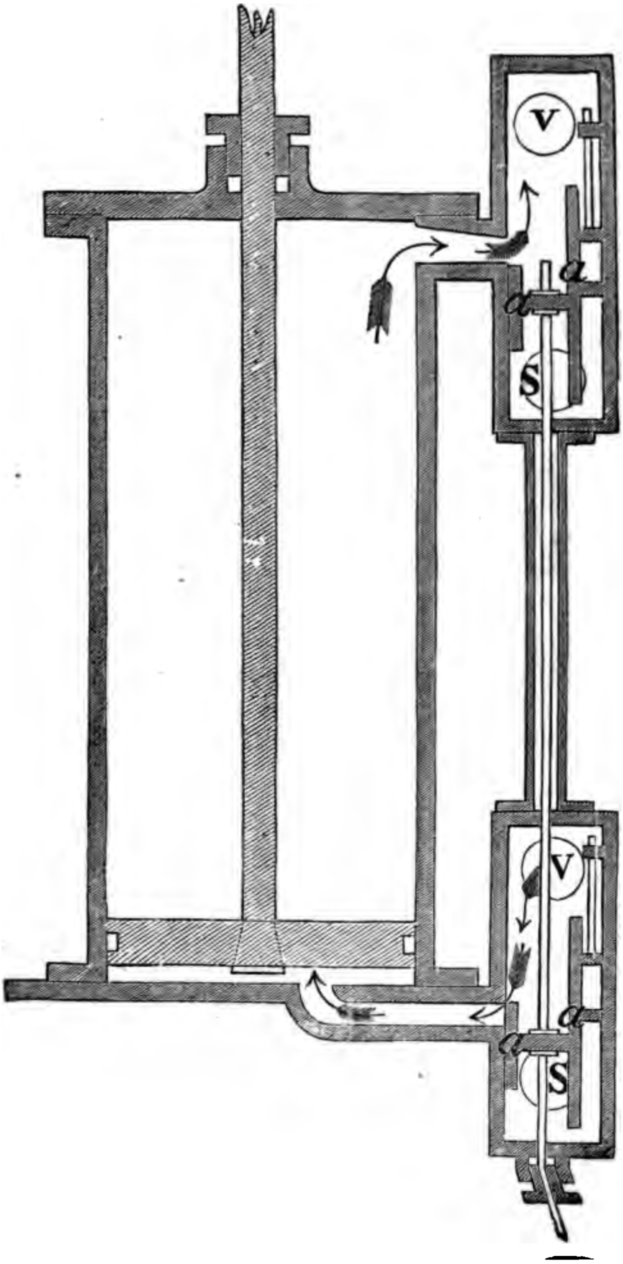
NOTE.—Those passages marked S are the openings of the pipe which convey the steam from the upper to the lower nozzle or Valve-chest, and the passages marked V are the openings of the pipe which convey the steam to the condenser.

Diagram No. 16.

SECTION OF A STEAM ENGINE CYLINDER, WITH D SLIDE VALVES, ROCK SHAFT, CRANK, AND ECCENTRIC.

Diagram No. 17.

AN ENLARGED SECTION OF STEAM ENGINE CYLINDER WITH D VALVES.



NOTE.—At the bottom part of the Valve in this illustration the letters S and V should be transposed, so that the vacuum side of the Valve will be at the extremity at both top and bottom.

they work, yet I do not confine my claim thereto, as they may be slightly increased, and even less extensive Valves may be used, varying the other parts accordingly; but I have found there is little advantage to be derived when the Valves are not at least equal to twice the length of their ports, measured in the direction of the movement of the Valves. What I claim is, the mode of constructing the Valves of Condensing Steam Engines, when worked by eccentrics or cranks, whereby the exhaustion ports may be fully opened at the completion of the stroke, and whereby steam may be cut off, and more effectually worked expansively by the same Valves as above described."

This description of the "lapping" and traversing of the D Slide Valve, given by Mr. Bould in the specification of his patent, is evidence that if he was not the first inventor of the process, he was in advance of many Engineers of the day; and there can be no doubt that the publication of his specification supplied information which has enabled other Engineers to improve the working of the Steam Engine.

There are other descriptions of Cylinder Valves than those described, such as the Long D, the Trellis, the Piston, and the Single and Double-drop Valves, all of which are one and the same in principle. An Engineer acquainted with the principle of the Valves here explained will have no difficulty in understanding all the various Valves in use, and the modes by which they are worked. For the latter, the eccentric movement is in most general use; but there are also the movements of the cam, the tappet, the crank, and the lever.

THE CORNISH OR DOUBLE-BEAT DROP VALVE.

This description of Valve derives its name from its first being adopted in Cornwall. In that locality Steam Engines are largely used for mining and pumping purposes, and the piston of the Engine travels at a slow rate. A Valve, therefore, that could be closed or opened on the steam and exhaust sides separately, became a useful and important improvement over the old "hand gear" Single Valve, and also over the Three-port, or D Slide Valve.

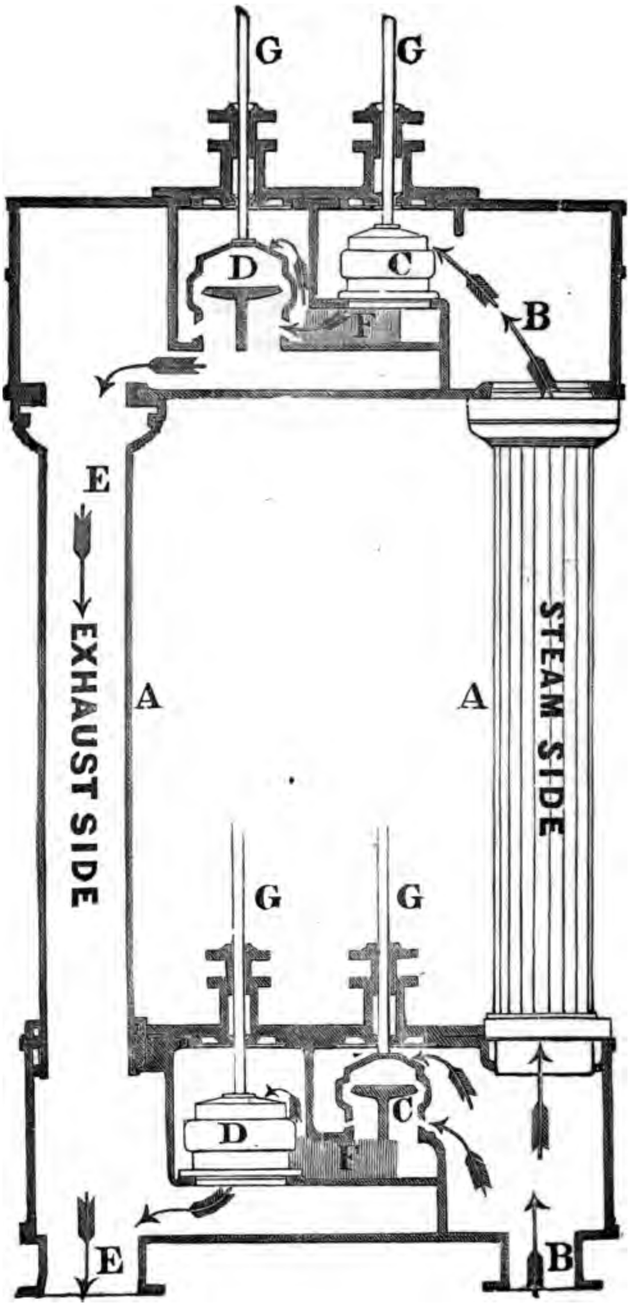
Diagram No. 18 is a vertical section of a pair of side pipes of a

Steam Engine cylinder, with Valves of the Cornish construction. These last are shown in their respective positions, both on the steam and exhaust sides. The steam from the boiler enters the steam-chest at B, and takes the direction as shown by the arrows at C, entering both at the under and top sides of the Valve to the piston: F F are the thoroughfares to the cylinders. The bottom Valve C being open, allows the steam to enter the thoroughfare F; and the top Exhaust Valve D being also open, allows the used steam on that side of the piston to escape from the cylinder through the thoroughfare F, and through the Valve D into the exhaust pipe E E, and thence to the condenser. The bottom Exhaust Valve D being closed, prevents the steam from passing from F to D; and the top Valve C being also closed, prevents the exhaust steam from the cylinder F passing through C to B. At each stroke or reversal of the piston, the Valves change their position: G G G G are the Valve spindles connected to a rod with arms, as shown by Diagram No. 19, which is a side elevation of the cylinder and side pipes.

In Diagram No. 19 the cylinder I, and the Valve-chests K K, together with the tappet P, and the projecting arms M M, attached to the main upright Valve rods, are sufficient to show the parts necessary to explain the action of the Cornish Valve. The tappet P is fixed to a horizontal shaft under the floor of the Engine house, and makes exactly the same number of revolutions as the crank shaft. Each Valve has its separate spindle and arm, and separate rod and tappet also. As the tappet shaft P revolves, the full side of tappet P comes in contact with small pulleys fitted to the bottom of the Valve rods, and raises the Valves from their respective seats, according to the requirements of the case. Pulleys are used to cause as little friction as possible in the working of the tappets, which are four in number. The advantage in the opening and closing of the Valves by this method consists in so forming the tappets that the steam Valve can be opened and closed at any required distance the piston may have travelled—giving every facility for cutting off the steam, either gradually or suddenly, according to the description of work on the Engine, or the amount of expansion of the steam required.

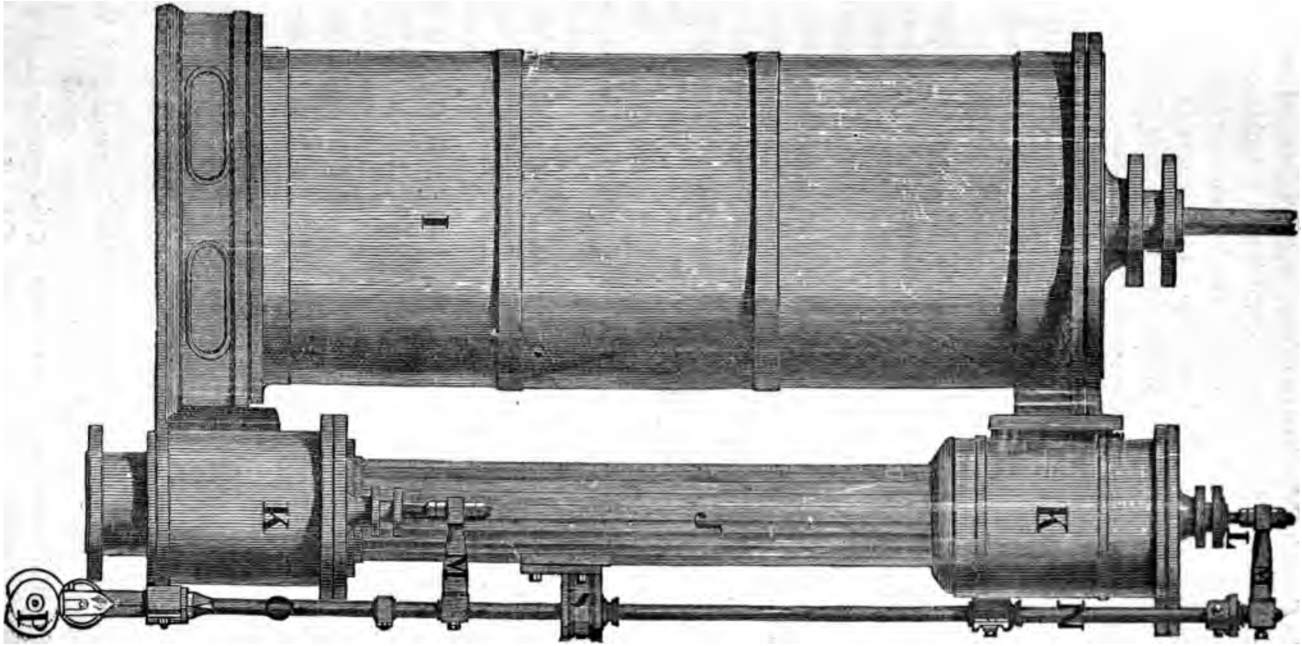
The Exhaust Valves have also separate tappets on the same shaft; and as they are required to keep open the Valve nearly the full length of the stroke, the tappet is so formed as to cause the Valves to open and close at the most advantageous times.

Diagram No. 18.



SECTION OF A STEAM ENGINE CYLINDER SIDE PIPES, WITH CORNISH DROP VALVES.

Diagram No. 19.



SIDE VIEW OF CYLINDER AND SIDE PIPES, WITH CORNISH VALVES AND REVOLVING TAPPETS.

The Cornish Valve has two faces, or two inlets and outlets, as shown by section of Valve C and D, which give a double opening for the ingress or egress of the steam. It also admits of the Exhaust Valve being raised much higher from its seat, so as to give a free and uninterrupted passage to the condenser. This form of Valve possesses a decided advantage over the Single-faced Drop Valve. When the tappet has raised the Valve to the highest point required, it again falls to its seat by its own weight, and is kept there by the pressure of steam, until the tappet again comes in contact with the pulley at the foot of the Valve-rod, and is again raised.

On reference to the illustration it will be seen that the Exhaust Valve is larger than the Steam Valve, which, coupled with the extra height the Exhaust Valve rises from its seat, give free egress for the steam to the condenser. With Double-beat Valves the steam can be kept longer on the piston before exhausting, because the Exhaust Valve being large, allows the steam to escape much quicker than the Three-port, or D Slide Valve. When Drop Valves are used, and worked by a revolving shaft and tappets, they are much easier to get at by the Engineer than the Three-port or D Slide Valve. Each tappet being separate, the Valves can be regulated separately, either by the shaping of the tappet's circumference, or moving it partially round the shaft, so as to give more or less lead on the steam or exhaust sides of the stroke; or to take off or keep on the steam in the cylinder, according to the load or pressure: thus enabling the Engineer to economise fuel by working his steam more expansively when less power is required. In this form of Valve there is often one defect: the tappets are so formed as to allow the Valve to drop upon its seat too suddenly, causing much wear and noise. Were the tappets made to allow the Valves to be steadied on their seats, the Valves would keep steam-tight much longer than the present construction.

IMPROVED SLIDE VALVE, WITH DOUBLE EXHAUST.

DIAGRAM No. 20 is a representation of the Improved Slide Valve, with double exhaust; S and B are the Valve chest, V the Slide Valve, P and D the steam-ways to each end of the cylinder, and E the *exhaust*. *It will* be perceived that the Valve has a considerable

amount of "lap," which will cut off the steam early, and thus work expansively, proportionate to the amount of "lap."

Diagram No. 21 is another representation of the same description of Valve, except with less "lap." The same letters denote the same parts as in No. 20.

It is well known that when "lap" is applied to the ordinary Slide Valve, the exhaust interferes too early with the action of the Engine, and allows the steam to escape to the condenser or atmosphere before it has had full opportunity of giving out to the piston the whole of its impelling force. This defect prevents that economy of fuel which would otherwise be realised from the working of steam expansively.

With the Slide Valve of the ordinary construction, the exhaust has to be set forward exactly in proportion to the amount of "lap" applied over the other thoroughfare, where "lap" is sufficient to cut off the steam so as to work with any considerable degree of expansion, and thus secure a reduction in the consumption of coal. The exhaust interferes in the manner above pointed out. To remedy this, many Engineers have abandoned both the Slide Valve and the eccentric, and have adopted other forms of Valves, of a costly and complicated description ; but these complicated constructions require much nicety of management, and are liable to derangement.

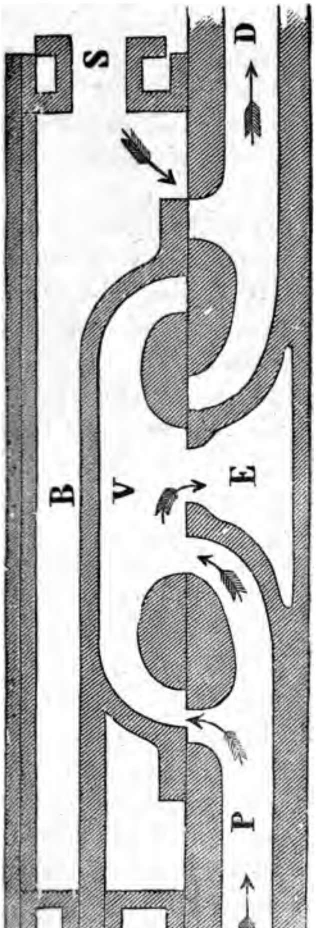
On reference to Diagrams Nos. 20 and 21, it will be observed that the steam passages are changed, both in the side pipes and in the Slide Valve, and that they have such a form given to them as to enable "lap" to be conveniently applied on the exhaust side of the Valve without any hindrance to the speed of exhausting the steam. It will also be perceived that two openings are provided for the steam to exhaust through, by which a quick exhaust is accomplished with half the traverse that is required by the old arrangement; and that, too, without such an enormous amount of lead being given to the exhaust in advance of the termination of the piston's traverse.

By reducing the traverse one-half, one half the amount of "lap" only is required. In the improved Valve very little "lap" on the exhaust side will be sufficient to confine the steam longer in the cylinder, so as to enable it to give out the whole of its expansive force upon the piston before it is discharged into the condenser or to the atmosphere. The old form of Slide Valve has been repeatedly tried to *accomplish this, with* "lap" on the exhaust side, but it

consequence of its long traverse, the amount of "lap" required was so great as to choke up the thoroughfares, and thus cripple the power of the Engine.

The short traverse of the Valve, under the new arrangement, has

Diagram No. 20.

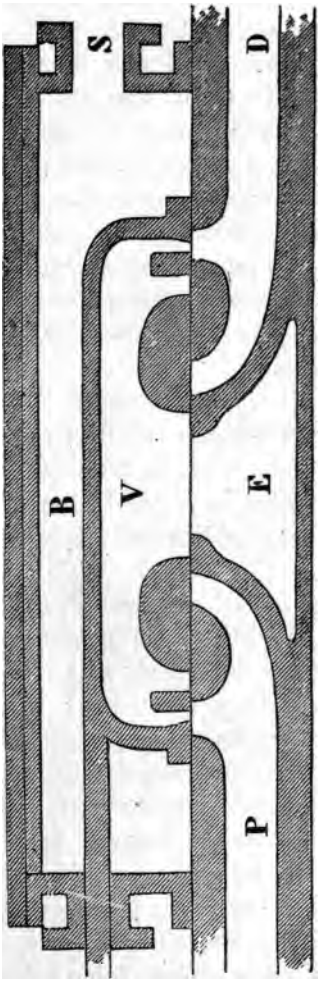


the advantage of admitting the steam on the piston slowly at the commencement of the stroke, and that, too, without a V being provided for the purpose. The steam can, therefore, be cut off much

sooner, and the expansive principle thus becomes more practically accomplished with Slide Valves of this construction.

The ordinary Three-port Slide Valve, having a Back Slide Cut-off

Diagram No. 21.



Valve worked by a second eccentric and rod, is shown by Diagram No. 22. By changing the position of the second eccentric the stea can be cut off at any desired point of the stroke. The Back Sl Cut-off Valve *is also frequently* made of the long kind, havi

...ves at each end,
... save steam room.
... and even three
...—the Back Slide
... sion of steam and
... of traverse of the

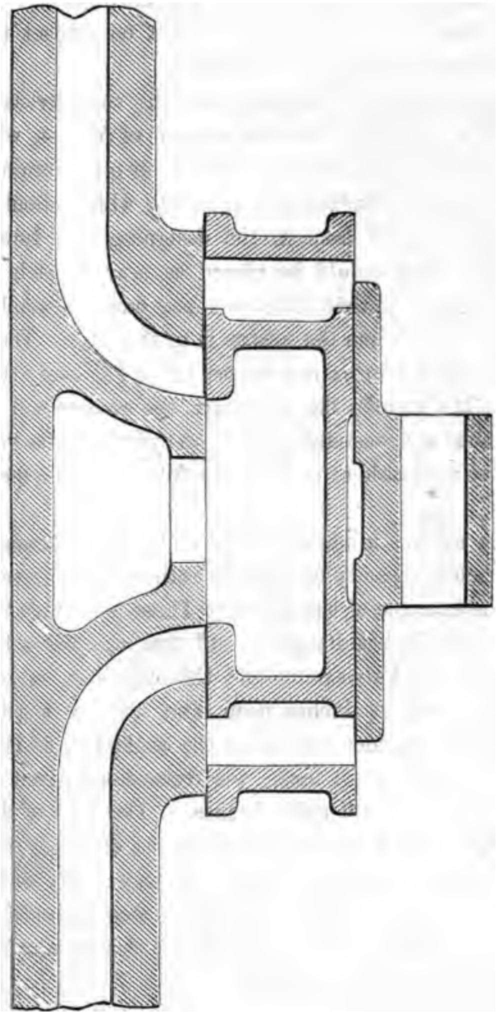
... which are given
... with this class of
... can be accomplished
... their own designs.
... which are peculiar, and
... advantages. For a full
... forms and arrange-
... is to lay down general
... judging of the merits of

... the designing of Valves
... for which they are
... discriminate. In
... Engines for general
... complied with, as far

... so that the full initial
... attained without violence.
... accelerating ratio, as the
... initial pressure of steam
... point of cut-off. Then.
... a sufficient part of the
... completely as possible.
... open a little before the
... should be completely open
... egress for the steam. It
... piston has nearly reached
... quickly and completely.
... none of the fresh supply of
...nser.
... Non-condensing and High-
... Engines. If, in a Condensing

ylinder, on the opening of the Exhaust Valve, the whole of the steam ontained therein should instantaneously rush into the condenser, eaving a perfect vacuum in the cylinder, then the Exhaust Valve

Diagram No. 22.



might be advantageously closed again at once. It is obvious, also, that the nearer perfect the vacuum in the cylinder, and the earlier may be the closing of the Exhaust Valve.

In this connection the different classes of Cut-off Valves naturally

suggest themselves; but these we will leave for a later part of this chapter.

Second.—The next principle to be observed in the construction of Valves is, that they should move with as little friction as possible, so that they may absorb the smallest amount of power. They will also require less lubrication, and will not be subject to as much wear,—all of which are highly important.

This question does not often receive the consideration which it deserves. For a calculation of the amount of friction which sometimes obtains with a Slide Valve, see chapter on Friction Diagrams.

Third.—Another fundamental principle which should never be forgotten nor departed from in the designing and construction of Valves, is this: they should be placed as near to each end of the cylinder as possible, so that the steam may have the smallest practicable space to fill, when the piston is at the end. The non-compliance with this condition is a source of loss at every stroke of the Engine; and the greater the departure, the greater will be the loss. This superfluous and wasteful capacity in cylinders is so very general, that it may be advisable to enter more fully into the question, and illustrate it.

There are several kinds of Valves and Valve Arrangements, in which this useless capacity occurs in a rather high degree. We will take, first, the example of the ordinary Three-port Slide Valve. Say the cylinder is 30 inches diameter, and that the piston has a stroke of 5 feet. Let the Valve-ports and thoroughfares, or passages, be 20 inches wide and $2\frac{1}{2}$ inches deep, and let the length from the Valve-face to the cylinder (including the angles) be 30 inches. The clearance, we will say, is 1 inch; the dimensions given for each of the thoroughfares $=1800$ cubic inches. The area of the cylinder $=706{\cdot}86$ inches$-6{\cdot}86$ inches sectional area of piston-rod (a small allowance, if passing through both covers) $=700$ inches. The thoroughfare, therefore, will equal $2{\cdot}57$ inches sectional area of the cylinder, and added to the one inch of clearance,$=3{\cdot}57$ inches, which is the whole fulcrum capacity.

Now, we will suppose the Engine to be working expansively. Say the initial pressure of the steam is 60lbs above the atmosphere, and that it is expanded to half an atmosphere. This will be $60+15=$ 75lbs$\div10=7{\cdot}5$lbs—being expanded to ten times its initial volume, *and to one-tenth its* initial pressure. The stroke of the piston being

60 inches, and the cut-off being at one-tenth, will be six inches traverse of the piston. As, however, we have seen that the fulcrum capacity amounts to 3·57 inches of the sectional area of the cylinder, and as this must be added to 60 inches—the length of the stroke—to give the total capacity which the steam must fill when the piston has reached the end of its stroke; therefore, the length will be 60+3·57 =63·57 inches; and as the point of cut-off must be one-tenth to admit the required quantity of steam, it will be 63·57 inches÷10= 6·357 inches. But, as the fulcrum capacity is 3·57 inches, therefore, the steam must be cut-off when the piston has traversed 2·787 inches. As, however, some amount of fulcrum capacity (clearance, &c.) is unavoidable, we will allow one inch, leaving the unnecessary and useless capacity=2·57 inches.

The diagram taken by the Indicator under the conditions above described, would show the cut-off at $\frac{1}{21}$ of the stroke. The expansion and the average pressure must be calculated, however, by the measure of the steam used.

Without presenting here the details of the calculations, it will be sufficient to state, that, assuming the vacuum at 13lbs, the average pressure on the piston will be 22·77lbs. But, when we make the deduction for the 2·57 inches of superfluous fulcrum capacity—the average pressure will be reduced to 20·07lbs, which is only 88 per cent. of the former, and is, therefore, a loss of 12 per cent.

The allowance here made of one inch sectional area of cylinder for the whole fulcrum capacity is quite sufficient. Cornish Valves have also the same objectionable feature of useless and excessive capacity.

It is not necessary to go into a detailed examination of these, or of any other kind of Valves, in connection with this part of the subject, as the principles and calculations here given will be a sufficient guide in the investigation of any and every kind.

It may seem superfluous to say—yet it cannot be too strongly represented—that whatever may be the proportionate loss of steam arising from this cause, the same proportionate loss of fuel is sustained; and this must be the justification for some little reiteration. During recent years, the question of Cut-off Valves has received—and deservedly so—very great attention. The variou forms and mechanical details are almost numberless, though t fundamental *principles on* which they are based—with respect

their action in regulating the steam in the cylinder—are simply reduced to four classes.

First.—There is the Slide, or any other form of Valve having a definite, fixed point of cut-off, accomplished by lap or otherwise. This, again, is divided into slow and quick cut-off—one very common plan of accomplishing the latter, being the Back Slide Cut-off Valve. There are many and various other methods of effecting the same object, which are often determined by the form of Valve, or other circumstances in the case.

Second.—The next class of Cut-off Valve is one having the cut-off adjustable by hand, and at any time, so that the Engine may be worked at any degree of expansion, according to the requirements or the judgment of those who have control.

Third.—The third class of Cut-off Valve has a connection with the governors, whereby the governors are enabled to shorten and extend the point of cut-off by a *slow* motion—usually a screw. This class of Valves, like the two preceding ones, requires a Throttle Valve, as the slow rate at which the cut-off is changed by the governors, does not enable them to exert immediate control over the Engine thereby. The action of the governors on the cut-off in this case is supplementary to the action of the Throttle Valve, as the governors cannot change the point of cut-off sufficiently quick to regulate the speed without the Throttle Valve. This class of Cut-off Valve is almost invariably constructed so that the point of cut-off can be changed by hand also, in order that the Engine-driver may start the Engine more easily, or otherwise adjust it, as circumstances may require.

Fourth.—The fourth and best class of Cut-off Valves, of which there are several varieties, are properly regarded as possessing advantages over all others. They have a variable cut-off, completely and instantaneously controlled by the governors, and require no Throttle Valve. The forms of Valves, and mechanical arrangements for obtaining this action, are numerous and ingenious.

It is done by Cylinder and Plug Valve, as in the Corliss Engine;—by the Piston Valve with the axial (commonly called the twist) motion;—by the Back Slide Valve with the link motion, as in the Allen Engine, and several other recent and slightly varying forms; by the back slide carried on the back of the ordinary Slide Valve by the pressure of the steam, and having an adjustable stopper freely *controlled by the governors*, as in WALKER'S Patent, made by

Messrs. Omerod, Grierson & Co., St. George's Ironworks, Manchester; by a sliding and tapering tappet; by the Balanced Valve of Mr. Wilson (Nasmyth Wilson & Co.), of Patricroft, near Manchester; and by some other arrangements.

The object of all, in this class, being to regulate the speed of the Engine by controlling the cut-off directly by the governors, and maintaining a uniform initial pressure in the cylinder—or within a definite limit of the boiler pressure, which will necessarily be somewhat different in different Engines according to the varying circumstances of each case. Some of the Valves here enumerated have other merits of a high order besides that of variable cut-off, and answer to all the conditions above defined of a good Valve.

In Horizontal Engines, the *position* of the Valve is worthy of consideration. It is very desirable that they should be so placed that any water which may accumulate in the cylinder by condensation, or otherwise, should escape freely into the condenser; or, in the case of the high-pressure cylinder of a Compound Engine, it should escape equally freely into the connecting pipes or receiver, where provision may be made for its separation from the steam, or re-evaporation by the application of external heat in the form of steam jacketing, or contact of heated gases after leaving the boiler furnace.

If one set of Valves only be used for both supply and exhaust—as is generally the case—a very convenient plan, adopted by some good Engineering firms, is to have them so placed that the bottom of the port should be on a level with the bottom of the cylinder. Another plan is to have separate Supply and Exhaust Valves. With this arrangement the Exhaust Valves are usually placed beneath the cylinder, and the Supply Valves at the top, though some Engineers place the latter at the side. This arrangement possesses some special advantages. The Supply and Exhaust Valves can be adjusted independently of each other—a feature which will be appreciated by Engine-drivers especially.

They have also another advantage, which does not arise from mechanical principles, but from the laws of the transmission of heat.

When the steam passes through the Exhaust Valve into the condenser, the surface of the iron at those parts loses a considerable amount of heat thereby; and when a fresh supply of steam is admitted through the same orifices, much condensation is the result. By having *separate Supply* and Exhaust Valves the condensation

will be much less, because the Supply Valve has not been cooled to near the same extent, as the steam, during the period of exhaustion, has there been comparatively quiescent; and, as it is one of the simplest facts in connection with the laws of the transmission of heat, that the amount of cooling under such circumstances will be determined by the quantity of aqueous vapour coming in contact with the surface of the metal, and rapidly passing over it, thus carrying away the heat of the metal by convection; wherefore it may be assumed that economy will result from such arrangement.

The multiplicity of mechanical appliances for giving motion to the various classes and kinds of Valves here generalized is such, that a detailed description of them, even if possible, would be too lengthy to come within the limits of this work, so that the reader must in this view be guided by the principles enunciated herein.

CHAPTER IX.

INDICATOR DIAGRAMS.

THE variety of Indicator Diagrams and descriptions of the same, of which this chapter consists, are given to illustrate the different arrangements and conditions for utilizing steam within the cylinders of Steam Engines. Examples will be found from Engines of almost every form and speed and construction of valves. In each case the facts are stated in connection with the diagrams, which will enable the reader to judge as to the merits or demerits, and, by comparison, show conclusively the advantages to be gained by a knowledge and use of the Steam Engine Indicator. Many examples from Compound Engines will be found amongst the number, some of which possess peculiar interest.

The descriptions which accompany the diagrams, and the analytical examination of some of them, render it altogether superfluous to say more here. Every Diagram may be relied upon as an exact copy of the original, both in form and size; and every fact stated we have examined ourselves, or obtained on the best authority. It may be well to remark that these diagrams are numbered consecutively, and wherever in the course of the chapters on "Compounding," or "Steam Jacketing," *diagrams* are referred to, they will be found amongst this number.

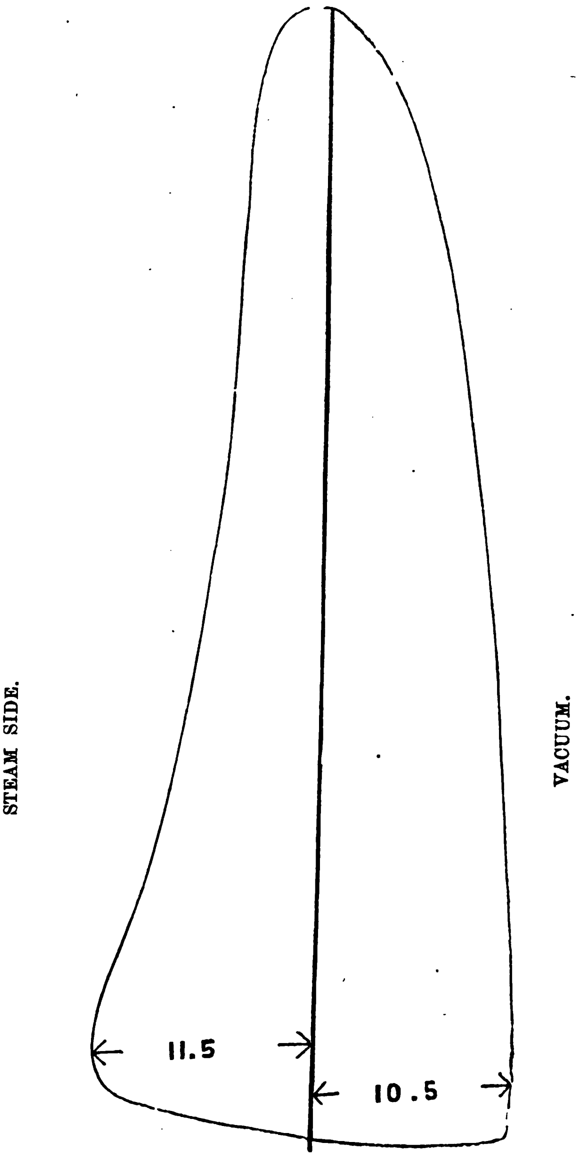
DESCRIPTION OF DIAGRAMS.

DIAGRAMS Nos. 1 and 2 are from the top and bottom of a Beam Engine, and have been taken since the year 1870. They belong, unmistakably, to the olden time. How many years the Engine has been working in this condition is altogether unknown. The boilers from which the Engine drew its supply of steam served for other purposes also, so that without the Indicator there was no means of knowing what quantity was being used by the Engine alone.

Hence the *careless indifference* which could permit an Engine to

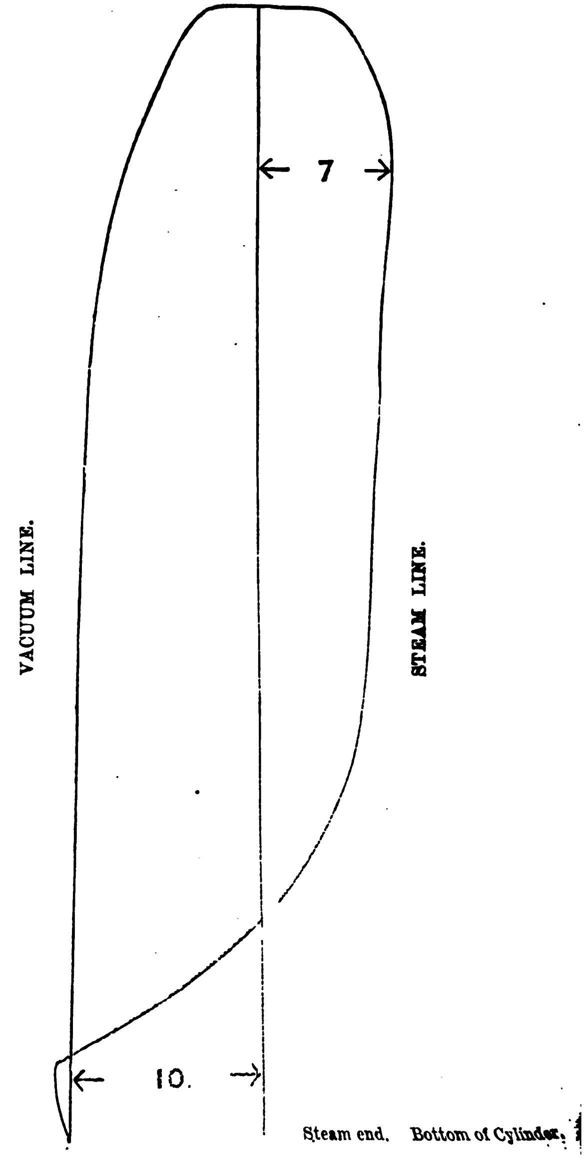
continue for an indefinite time in such a deplorable condition. If the quantity of fuel consumed, for the power obtained, had been known, it is fair to presume, that whoever had to pay the coal bills

Diagram No. 1.



would have had the Engine put into a better state with the leas delay possible. The application of the Indicator which shows th excessive waste of steam, shows also what was required to be done t

Diagram No. 2.



effect a very great improvement. Without going into calculations on the diagrams, it will be a moderate statement to say that two-thirds of the steam is wasted, or that the same power could be got from one-third the amount of steam, and from the same Engine, without any excessive pressure or strain.

It will be noticed in No. 1 Diagram that the steam is kept on nearly the whole length of the stroke, and maintains almost a parallel line. That the exhaust valve opens too late and closes too late. Indeed, the exhaust valve not only opens too late, but opens too little—not giving a sufficient orifice for the steam to escape rapidly into the condenser, and thereby preventing the vacuum in the cylinder attaining even a moderate amount until the piston has reached the end of its return stroke. Diagram No. 2 is about the same as No. 1 in respect to the exhaust, and, therefore, the same remarks apply. In other respects it is very much worse than No. 1, though that is bad enough. The practised eye will see at a glance that the valve opens too late for the admission of steam. Besides which, the opening is so gentle, and continued so long, that the highest pressure is reached at the termination of the stroke.

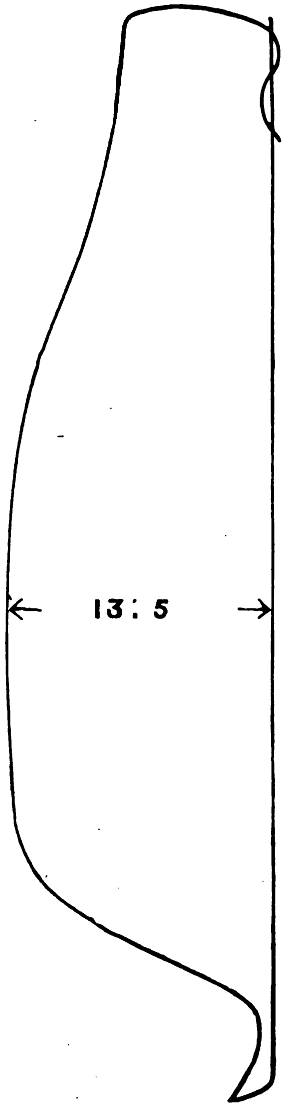
Diagram No. 3 is from a Non-condensing Engine (commonly called—though improperly so—High-pressure). It belongs to a class of Engines whose numbers are beyond computation, and is a fair and true representation of them. It is true that the steam is not continued at full pressure to the end of the stroke ; but the little merit it has there, is quite counterbalanced by the late admission. The good point in the diagram is that the back-pressure line is on the atmospheric line. In this case half the steam is wasted—calculated on the basis of a very moderate pressure.

Diagram No. 4 is somewhat better than No. 3. It has the same fault of having the admission of the steam continued too long, and consequently of having too much at the end of the stroke. There is another feature in this which was not seen in the other diagrams. The opening of the valve for the ingress of steam is too sudden, and causes the Indicator pencil to be carried past the point of steady pressure. The same concussive force will exist in the cylinder. With a Direct-acting Engine, having a quick speed, this will not be so dangerous.

To the uninitiated, Diagram No. 5 is a strange and curious figure. It was taken from an Engine belonging to one of the largest and

best known manufacturing firms in the country. The Engine (
not fall into this condition through ignorance, or the want of

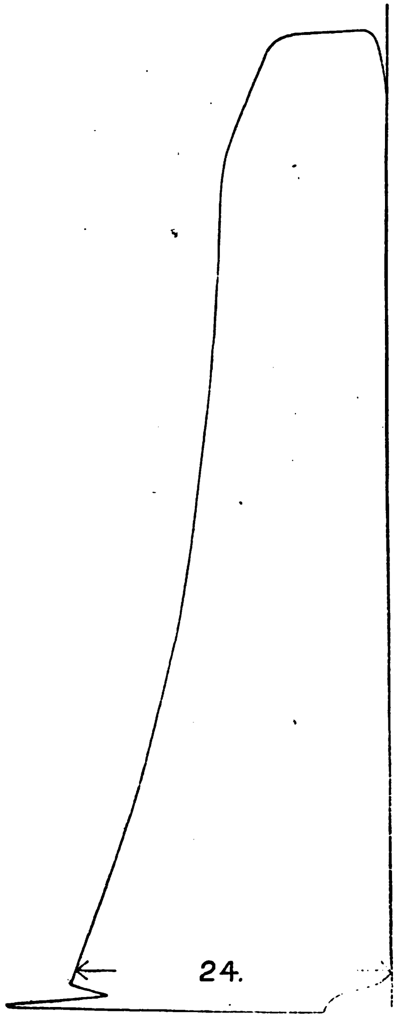
Diagram No. 3.



Indicator, for the firm had an Engineer of great ability. '
Engine was in *a satisfactory* state only a few days before this diag

was taken. The Engineer was told that something was amiss, inasmuch as the Engine was running a little under speed, and the stoker could not keep the steam up as usual. The Engineer applied the Indicator, and then saw at a glance that some part of the valve

Diagram No. 4.

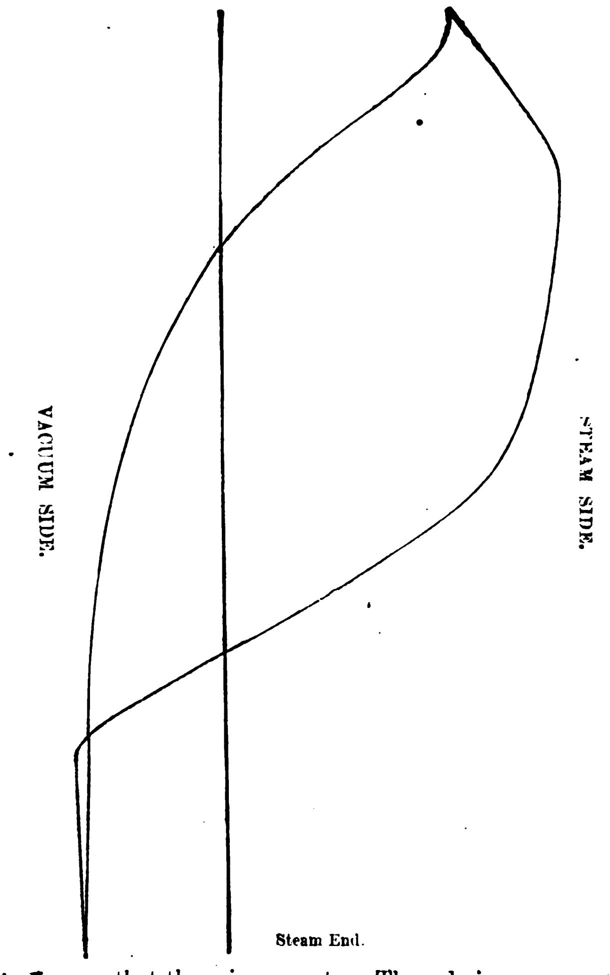


gear had given way, causing the steam to be too late in its admission. A very short time sufficed to discover where the fault lay, and not a *long time to correct it.* If this had occurred at some place where

the Engine is supposed to be kept right by some Engineer coming once in twelve months to indicate, it may easily be seen that a very great loss would have been sustained.

Diagram No. 6 is from a Non-condensing Engine. It is almost as good as can be produced. The terminal pressure is on the atmo-

Diagram No. 5.

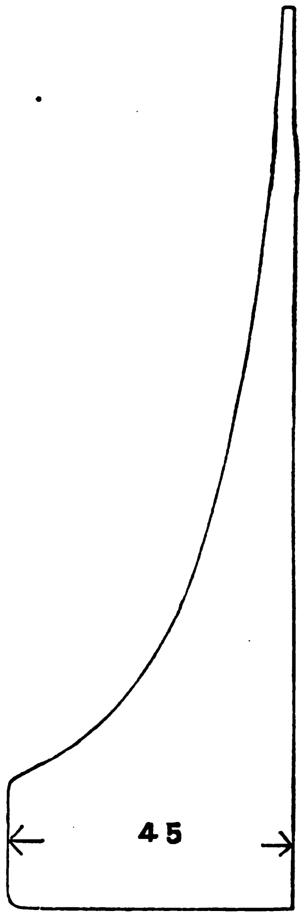


spheric line, so that there is no waste. The only improvement which could be made, would be to close the exhaust valve a little earlier, so as to secure a little compression, or what is generally called cushioning. The back pressure line being on the atmospheric line

is as good as is practicable. The valve is a back slide cut-off, and, as will be seen, acts admirably.

Diagram No. 7 is from a Horizontal Engine of five feet stroke, thirty inches diameter of cylinder, and forty-two revolutions per minute, making 420 feet per minute speed of piston. The Engine

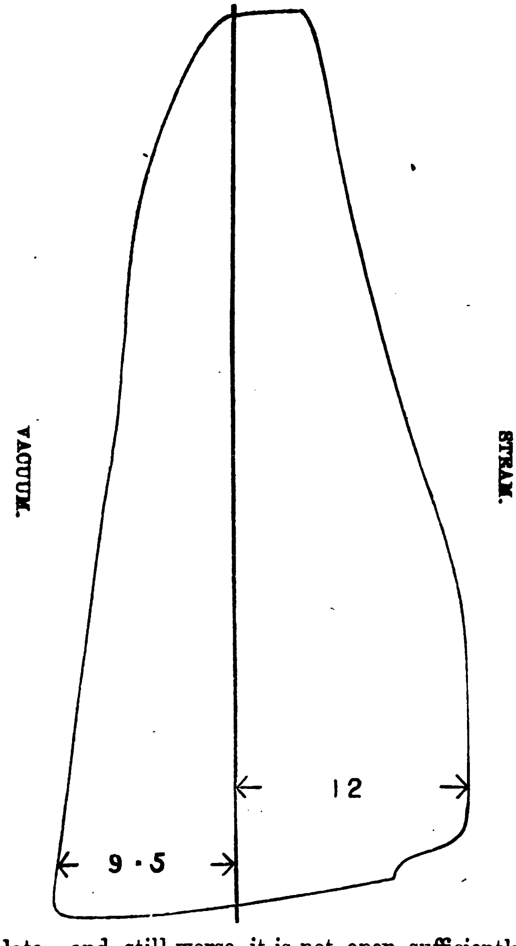
Diagram No. 6.



was made by an engineering firm of high repute. Whether it had ever been in a better state than this Diagram shows, the manager of the firm to whom it belongs did not know. It shows clearly that it *is not* wise or safe to trust simply to the reputation of the maker *for its proper* and economical working. If we suppose that the

Engine had been right originally (a very doubtful point in this case), it was still very important to indicate frequently, and adjust whatever was thus shown to be wrong. This Diagram is bad in every respect. The steam is rather too late at the commencement of the stroke, and is continued too long. The exhaust valve opens and

Diagram No. 7.



closes too late; and, still worse, it is not open sufficiently to permit the steam to pass freely into the condenser, causing the vacuum to be very bad, and only a small amount even at the termination of the exhaust line. Another cause may operate to produce this defective vacuum, viz.:—leakage of steam through the valves, or piston, c

both. This Engine ought to drive the same power with one-tl
the quantity of steam, and consequently with one-third the amo
of fuel.

Diagram No. 8.

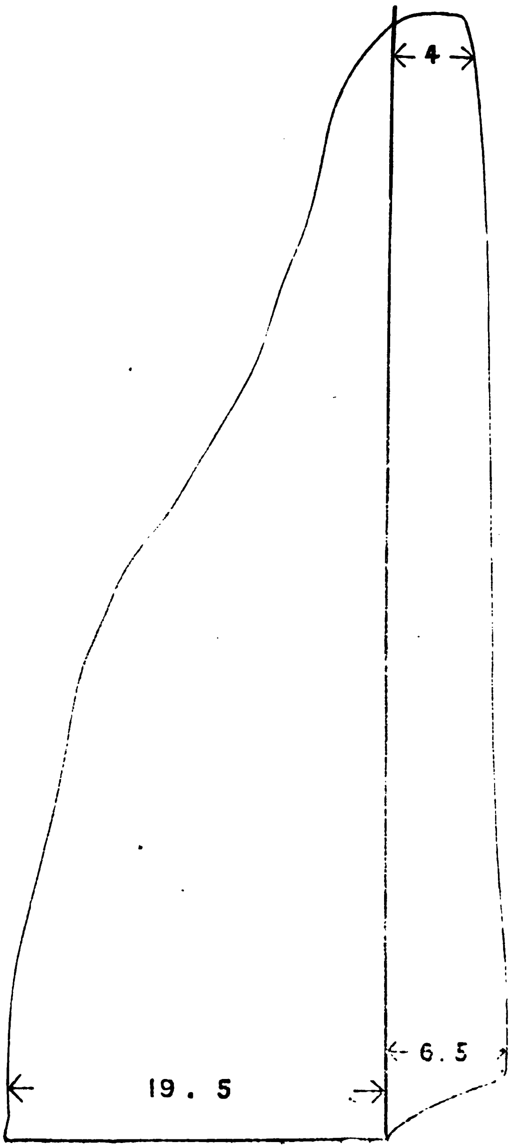


Diagram No. 8 is somewhat similar to No. 7 in its general features; and, like it, has simply a three-port slide valve. The special point observable in this diagram, is the small amount of vacuum. At the time of taking the diagram the vacuum gauge attached to the condenser showed 12lbs. This great defect in the vacuum was caused by the piston being bad (though quite a new Engine) thus permitting the steam to pass rapidly to the exhausting end of the cylinder. Every pound of steam passing the piston in this way is equal to a loss of 2lbs of pressure at least, but probably much more. Now, without touching the question of expansion, let us see what is the loss in this case simply arising from leakage of the piston. The vacuum shows an average of 5·5lbs; whereas, even with the rather high terminal pressure which will obtain with valves in the present form, the vacuum should be (to put it low) 10lbs. The difference between 5·5 and 10lbs being 4·5; and, as this must be doubled, as just shown, then the actual loss amounts to 9lbs average pressure per inch—being equal to one-third of the steam wasted, or only two-thirds of the power obtained from it.

Diagram No. 9 is from a Beam Engine having Cornish valves—in Lancashire frequently called stamper valves. The figure is of a very good general form. The opening and closing of the exhaust valve is excellent, and yet the amount of vacuum is deficient. The point particularly prominent in the diagram is the admission line, which is carried above the point at which it can be sustained by the mere pressure of the steam. This is caused by the valve being opened too rapidly, and the steam thereby rushing in with great violence. A corresponding vibration was communicated to the beam as appears on the diagram. The valve was only one day in this condition, as it was seen and felt that it would not be safe to continue. The Engine is one of a pair, with the cranks at right angles, and running at a speed of thirty revolutions per minute.

Diagram No. 10 is from a Beam Engine, made by Messrs. Petrie & Sons, Engineers, Rochdale. It has piston valves and the well known *twist* cut-off motion—the differential clutch gear—which moves the cut-off tappet forward or backward by the action of the governors. It is a comparison and a contrast with Diagram No. 9. For a light Beam Engine—such as it is—the diagram could scarcely be better. The steam, as will be seen, is admitted very gently, and cut off in such a way as to give a beautiful expansion curve, with a

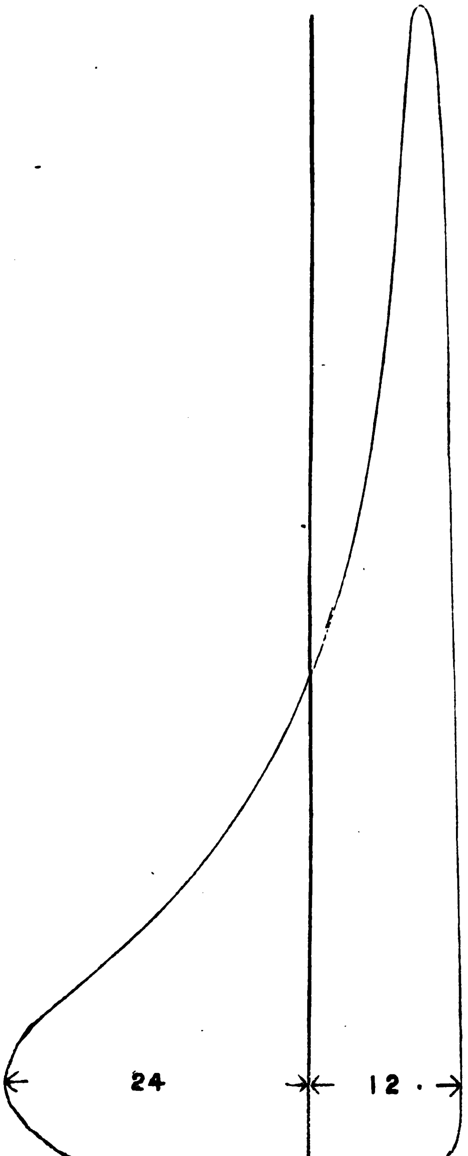
terminal pressure of 8lbs below atmosphere. Speed—30 revolutions per minute.

Diagrams Nos. 11 and 12 are from a Beam Engine, in the

Diagram No. 9.

neighbourhood of Bradford, which is driving a mill for the spinning and manufacturing of worsted goods. The Engine is in a very

Diagram No. 10.



unsatisfactory condition in almost every respect. The admission of the steam in No. 12 is very late. The opening and closing of the exhaust valves is too late at both the top and bottom (Diagram No.

Diagram No. 11.

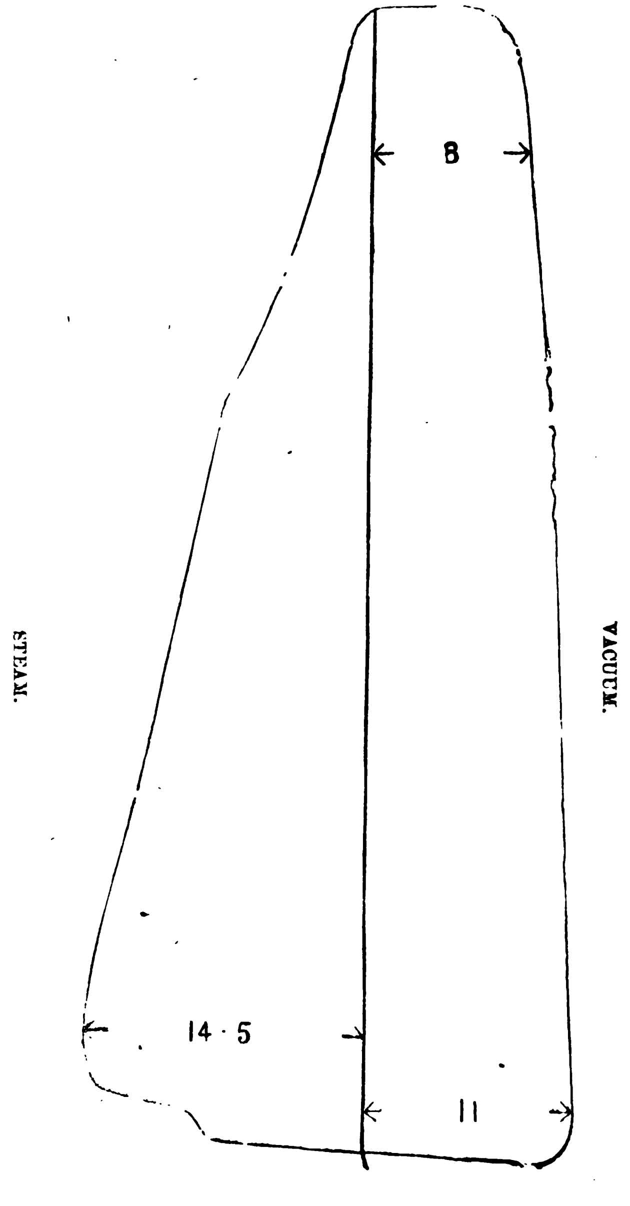
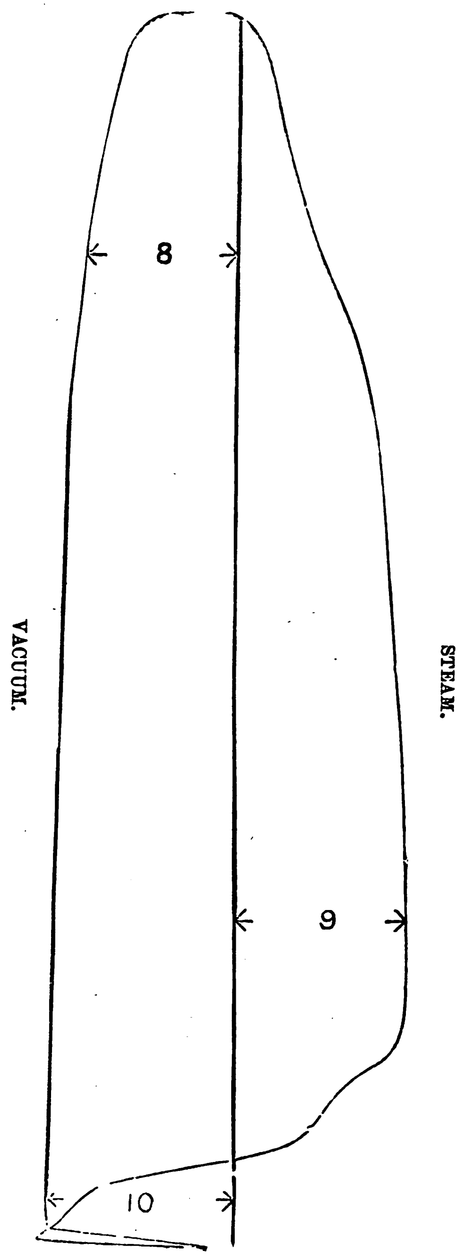


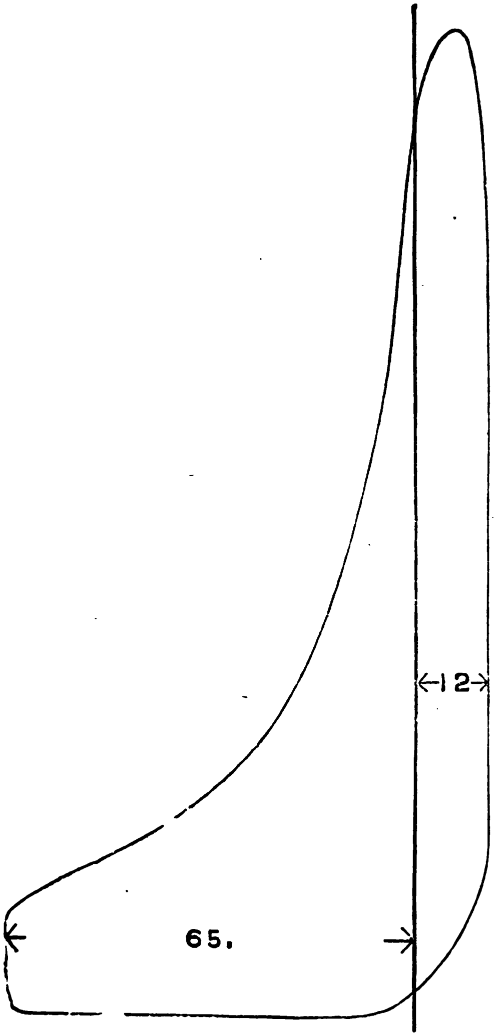
Diagram No. 12.



V

11 is from the top, and No. 12 from the bottom of the cylinder). The Engine is driving 49 I.H.P.; speed, 40 revolutions per minute; three feet four inches stroke, and 22 inches diameter of cylinder. The same power could be got from half the quantity of steam, even

Diagram No. 13.



with the same Engine, if the valves were of a proper kind and properly adjusted. Small firms usually work at a great disadvantage *in the cost of* power—not because costly running is an inevitable

condition which appertains to a small concern—but it is generally supposed not to be worth the trouble and expense of attending to such things—that, indeed, it would not pay to do so. In many cases little expense would be incurred, and a great saving effected.

Diagram No. 14.

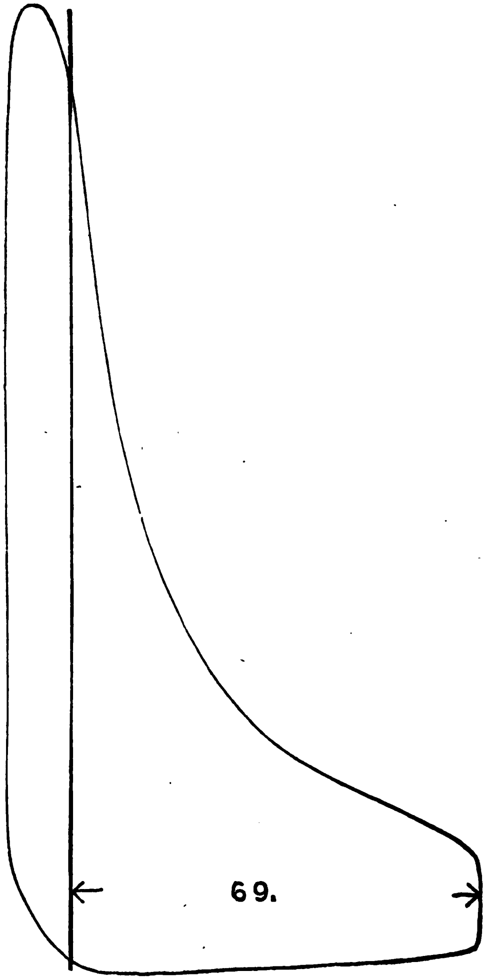


Diagram No. 13 is from one of a pair of Horizontal Engines, at a Cotton Spinning Mill. They are five feet stroke, 26 inches diameter, and run at 48 revolutions per minute,—making 48

feet per minute speed of piston. Valves—Back Slide Cut-off. The diagram is a very beautiful one. The admission shows that the maximum pressure is attained without violence. The steam line is parallel, showing it to be maintained at a uniform pressure to the point where the cut-off commences. The cut-off is quick, as is evident from the diagram—the true expansion line commencing almost immediately after the admission valve begins to close, and continuing to the point of intersection with the atmospheric line, where the exhaust valve begins to open. The closing of the exhaust valve forms a beautiful compression curve.

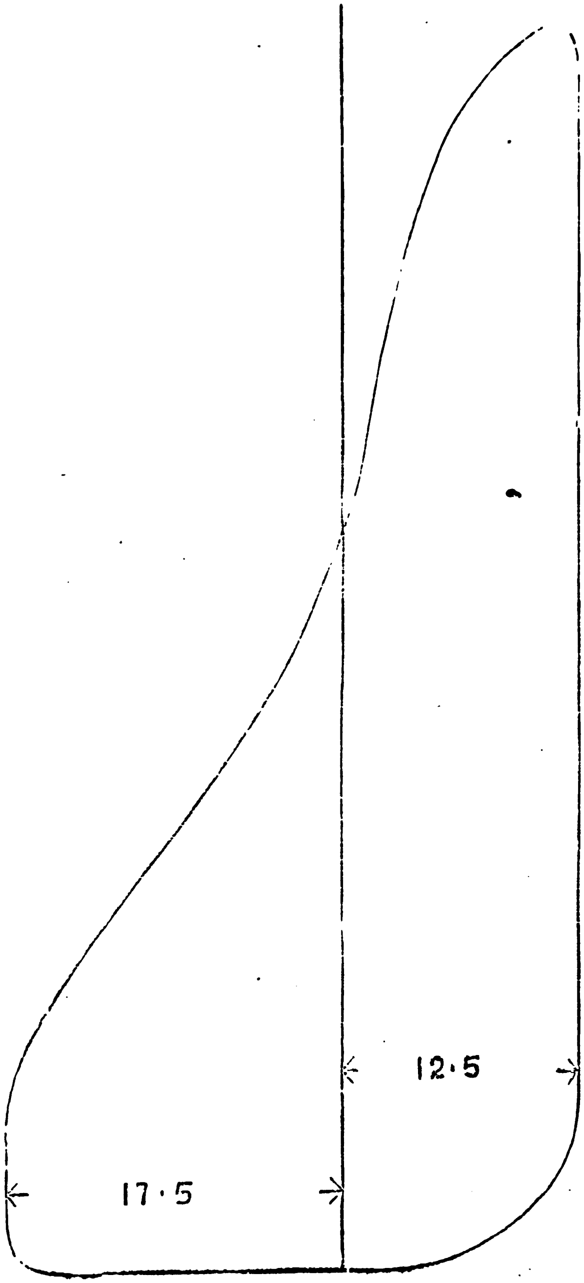
Diagram No. 14 was taken by the Richard's Indicator, from the same tap, and under exactly the same conditions, except that the pressure had fallen a little, both in the cylinder and in the boiler, when the former was taken. The admission line in Diagram No. 13 rises at right angles with the atmospheric line, when it reaches the maximum initial pressure by gently rounding the corner into the steam line. Diagram No. 14 shows the admission line slightly receding from the perpendicular, the Indicator being unmistakably rather late in responding to the action of the steam.

An examination of the vacuum lines of the two diagrams at the point where the compression line begins, that is, the closing of the exhaust, shows the same characteristic phenomenon. The compression line is seen to commence earlier in Diagram No. 13 than it does in No. 14, proving the action of the Indicator to be much less affected by friction in the former than it is in the latter case.

Diagram No. 15 was taken from one of a pair of Beam Engines, belonging to Messrs. JOSHUA CRAVEN & Co., manufacturers, Thornton, near Bradford. They are of five feet stroke, 28 inches diameter of cylinder, and 30 revolutions per minute, and have long D Valves. It shows the Engine to be in almost the best possible condition. Indeed, it is difficult to conceive how better results could be obtained with the same initial pressure of steam. No part of the diagram can be altered with advantage—taking into account the speed and all other circumstances. It may truly be pronounced a perfect diagram of its kind.

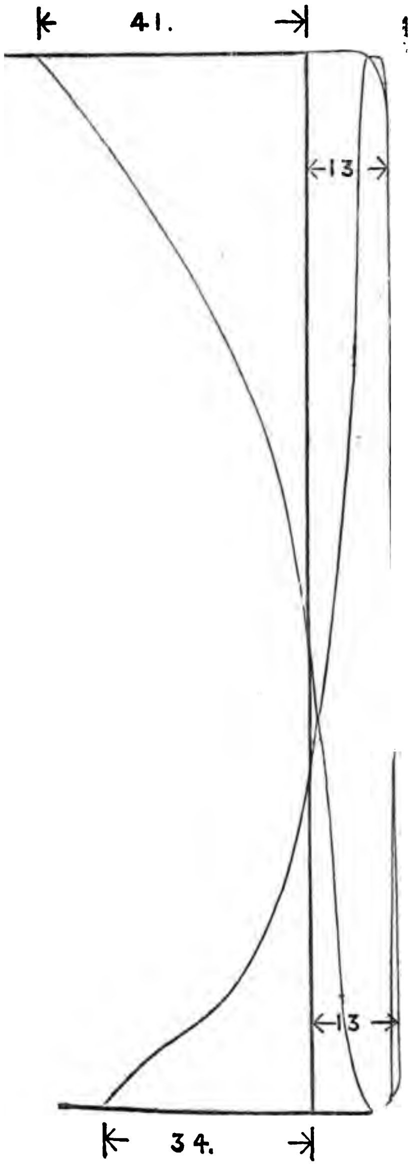
No. 16 is a double diagram, taken from a Horizontal Engine, 60 revolutions, three feet six inches stroke, and 24 inches diameter of cylinder. It has a piston valve with twist cut-off motion. The cut-off is not so complete and decisive as we generally find

Diagram No. 15.



in valves of this description. The opening of the valve is too sudden, as shown by the diagram, and felt, when standing by the Engine, as plainly as it is seen on the diagram. The weight of steam

Diagram No. 16.



is much greater at one end than the other. This is a common occurrence which can only be remedied by adjusting to the requirements shown by the Indicator. The vacuum is excellent.

Diagram No. 17, if presented without any explanation, would be as incomprehensible to 99 per cent. of Engineers as Mexican hieroglyphics. A description of the circumstances in which it is produced will, however, soon enable the reader to understand it. The Engine is for compressing air and sending it down the shaft of a coal pit to drive Engines at the bottom, instead of using steam. The speed is only six to seven revolutions per minute. The steam and air cylinders are both in one line, with the piston-rod passing from one to the other. The steam cylinder is seven feet stroke, and three feet nine inches diameter, and has Cornish valves. It is a Non-condensing Engine. It will be noticed that the pressure at the commencement of the stroke is 26lbs above the atmospheric line, and runs gently down to 14lbs, after which it again suddenly rises to 25lbs pressure, which is maintained to the end of the stroke. This arises from the fact, that at the commencement of the stroke the pressure in the air cylinder is at zero, but as the piston advances the pressure of air increases gradually until it has attained its maximum of 40lbs, above which it cannot rise. This pressure is reached when the piston has traversed two-thirds of its stroke. The momentum of the fly wheel, however—the speed of which is rapidly accelerated at the early part of the stroke—carries the piston past the point where the maximum pressure in the air cylinder is first reached, until, when it arrives at the point where the notch in the diagram occurs, the momentum has been expended, and it comes to a dead stand, and only moves forward the remainder of the stroke by the whole cylinder being filled with steam of 25lbs pressure above atmosphere. The back pressure also is a source of great loss. To have 11lbs of back pressure during half the stroke, is a state which alone would alarm any one who had the coals to pay for. Yet, this is only a small loss in comparison with the steam side of the diagram. In this case there are difficulties in the way of coming to an accurate estimate of the amount of loss, but it is undoubtedly very great.

A precaution has been urged at page 93 in respect to placing the cylinder taps so near the end of the cylinder (if not put in the cover) that the piston shall not pass or cover the orifice when at the end of its stroke.

Diagram No. 17.

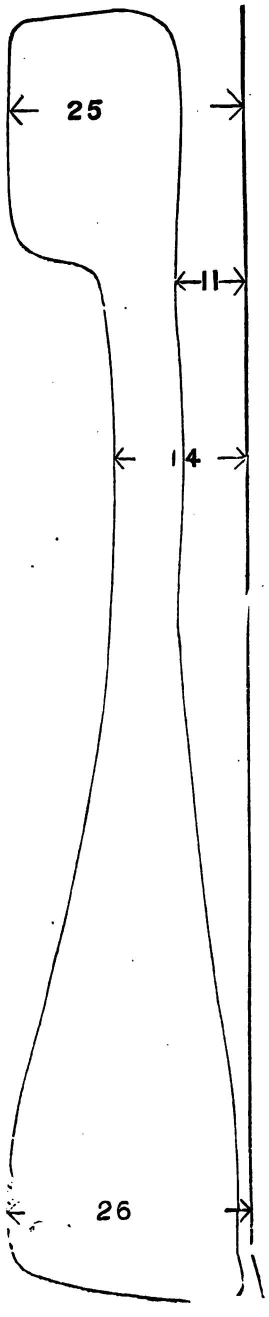


Diagram No. 18 is an example of such an error. In this case we will leave the reader to his own ingenuity in the further deciphering of it. It is clear that no reliable judgment can be arrived at from such a diagram.

Diagram No. 18.

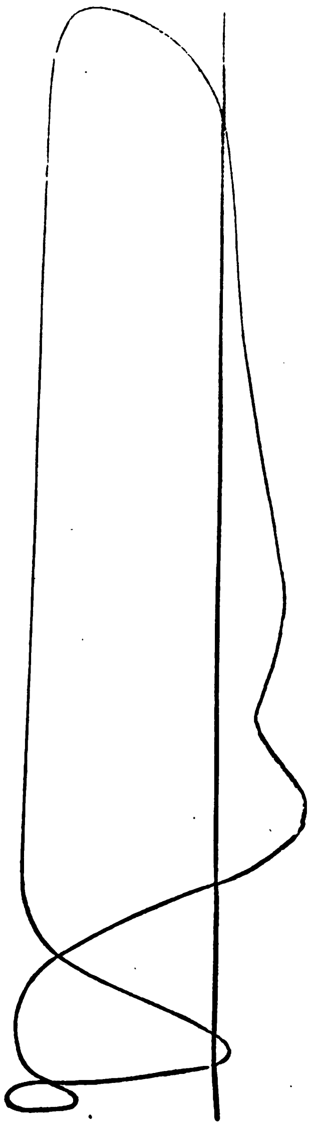
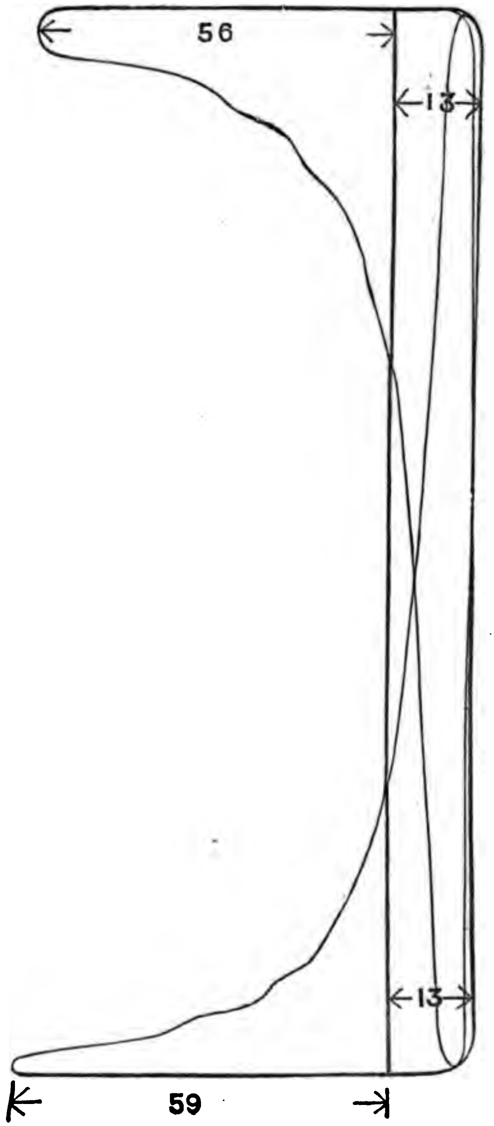


Diagram No 19 is a double one, from a Corliss Engine. Th
one from which this diagram is taken is a Horizontal Engine, a
most of the Engines with Corliss valves are. It is four feet strok
two feet diameter, and runs 52½ revolutions per minute. The valv

Diagram No. 19.



are called " INGLIS & SPENCER'S Patent Improvements of the Corliss Valve," and are made by Messrs. HICK, HARGREAVES & Co., Bolton. On examination of the diagram it will be seen that the valves move with perfect scientific accuracy. Though the Engine runs at a fast speed, yet almost the full boiler pressure is admitted on the piston in the most gentle manner possible. Any consecutive number of diagrams will show an equal degree of perfection. The cut-off is equally regular and smooth in its action on the steam. At one end of the cylinder the steam is shown to be cut off at about one-twentieth and at the other end about one-fortieth of the stroke. The terminal pressure of the one-twentieth cut-off is 8lbs below atmosphere, and the terminal pressure of the one-fortieth cut-off is 9lbs below atmosphere. This merits a few words of explanation. If the piston, at the end of its stroke, left no space between it and the cover, that is, if there were no clearance and no capacity whatever beyond that which is left by the advancing piston, then the difference in the terminal pressure would assuredly be much greater than is found in this diagram. Such, however, is not the case. There is the amount of clearance and the ports, &c., and these, in addition to the distance moved by the piston, constitute the capacity which the steam occupies at the point of cut-off. There is also the condensation on the admission of the steam which affects the terminal pressure to an extent something difficult to determine. The initial condensation is relatively greater as the cut-off is earlier, and the ratio of expansion is greater, as will be shown hereafter in connection with other diagrams. The cylinder being jacketed and supplied with steam at the boiler pressure, a large portion of the condensed steam is re-evaporated,—whether all of it, cannot here be determined. Had the points of cut-off been one-fifth and one-tenth, then the relative difference in the terminal pressures would have been much greater, because the clearance in the latter case would have borne a smaller proportion to the whole capacity of steam at the time of cut-off. This part of the diagram will now be clearly understood. The fact of one end of the cylinder having the steam cut-off half the length of the other end is very simply explained. In these Engines there is no throttle valve—the speed being regulated by the admission valve, which slips back instantaneously by the action of the governor when sufficient steam has entered *the cylinder for* its requirements. With the slight

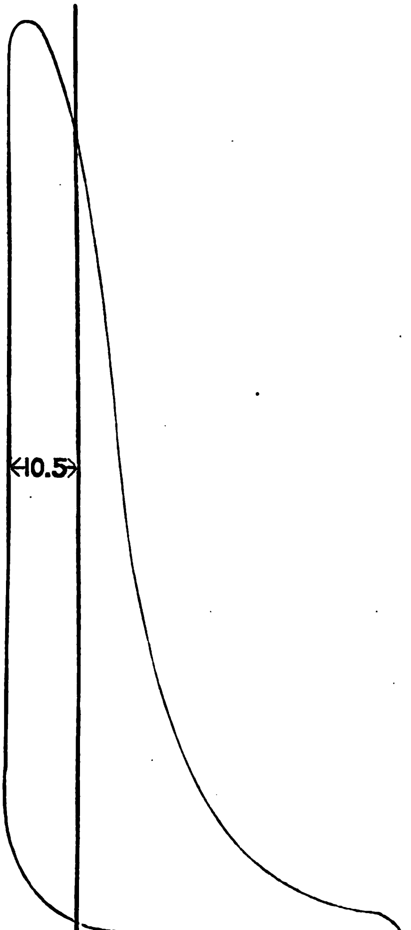
change of speed the governor will cause the cut-off to be earlier or later as the speed may require. The very next diagram taken might have the relative points of cut-oft reversed. Whatever may be the point of cut-off, the maximum pressure in the cylinder will be about the same, and will bear a fixed relation, according to circumstances, to the boiler pressure, whatever that pressure may be. In this instance, the boiler pressure is 60lbs. The initial pressure in the cylinder usually ranges from 1lb to 4lbs below the boiler pressure for the time being.

This diagram shows a most excellent vacuum—almost the best attainable—which, with the speed of the Engine considered, proves the air-pump to be good, and the exhaust valves to be admirably arranged. The exhaust valves are shown to be full open to the end of the stroke, and then to close quickly. There is no cushioning here. The early cut-off and low terminal pressure are such, that the momentum of the reciprocating parts will be fairly expended on reaching the termination of the stroke for this moderate speed of Engine. A higher speed would necessarily require more compression. There is yet another feature in this diagram which deserves some attention. It will be noticed that the expansion line falls in steps or terraces. This is not in the least degree objectionable. The cut-off is so instantaneous, that this must be so with any instrument sufficiently sensitive to be reliable in other respects. The mean pressure will be got from it equally well. The opening is steady and regular—this perfect regularity being reproduced in any number of diagrams. This is an example of a very high degree of expansion.

Diagram No. 20 is from an Engine made by the Whitworth Engineering Company, Limited (Sir JOSEPH WHITWORTH & Co.), Manchester. It is a Horizontal Engine, two feet six inches stroke, eighteen inches diameter, and runs at 120 revolutions per minute. It is the famous "Allen Engine," the first one seen in this country, being brought from America by Mr. C. T. PORTER, and shown at the International Exhibition of 1862, The Engine takes its name from the inventor of the valve and valve motion. The valve may be described generically as a back-slide cut-off valve. The cut-off is completely under the control of the governor, and acts admirably—the Engine requiring no throttle valve. The arrangement for changing the position of the back-slide valve, by the action of the governor, is unique and beautiful. This diagram shows how

sweetly and smoothly the valves work. The diagram shows the terminal pressure very high for the early cut-off. The speed being very high, the steam, which on entering a condensing cylinder, is always condensed more or less, is here reconverted into steam as the

Diagram No. 20.



piston advances. The cylinder being steam jacketed, secures the more complete re-evaporation of the steam which had been condensed on its first *entrance*. Besides which, the diagram seems

to indicate a considerable amount of water carried with the steam into the cylinder, some of which, in this case, will be converted into steam as the piston advances. Hence, probably, the high terminal pressure. The expansion curve is a beautiful one for such a speed. The vacuum seems small for such a first-class Engine. But when we take into account the very high speed and the rather high-terminal pressure, it is clear that such a vacuum as some of the diagrams show is simply impossible. The opening and closing of the exhaust valves is nothing short of perfection. Indeed, the diagram is altogether a gem.

Diagram No. 21 is from a Horizontal Engine (a pair run together). Speed, 48 revolutions; stroke, five feet; and diameter of cylinder, 28 inches; with Cornish valves. The admission line shows the steam to reach the full initial pressure before the piston moves. This is quite safe and useful for a Horizontal Engine at this speed. The cut-off is not so decisive as many other Cornish valves show. This is determined partly by the shape of the tappets, and partly by another fact in this case. These Engines have got very large ports and valve box, so that a considerable capacity has to be filled with steam besides the capacity left by the distance which the piston has moved at the point of cut-off. Another, and still more potent cause of the indefinite and undistinguishable cut-off, is one which appertains to a very great number of Engines. It is this: The amount of the opening of the valve is not sufficient to admit as much steam as will maintain the initial pressure when the piston begins to move with accelerating speed. Hence there will be a fall in pressure, even though the valve may remain equally open, and when the valve does close, there is no falling of the pressure in the cylinder, as shown by the diagram. It is impossible to determine at what point the valve really closes. To assign to each of the three causes here described, its exact value in its influence on the expansion line, would require more exact data than we have at hand.

Diagram No. 22 is from the top of one of the large Beam Engines of Messrs. Eccles Sharrock, Brothers, and Company, Over Darwen, and which were made by Messrs. W. and J. Yates, of Blackburn. They are seven feet six inches stroke, four feet three inches diameter, and run at twenty-eight revolutions per minute.

The Valves are Cornish, and the tappets, which are movable, *have four lifts,* thus providing for four points of cut-off. Diagram

No. 23 was taken at the same time, or immediately after, from the same tap, by the RICHARDS' Indicator.

Diagram No. 21.

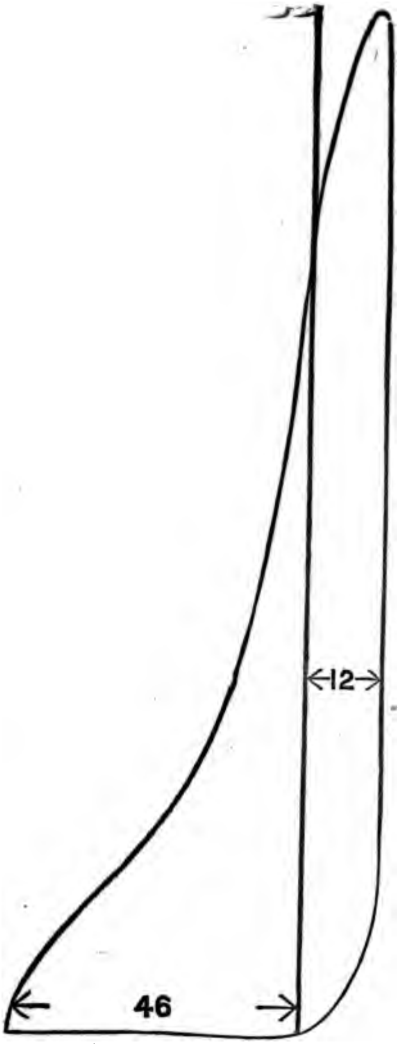
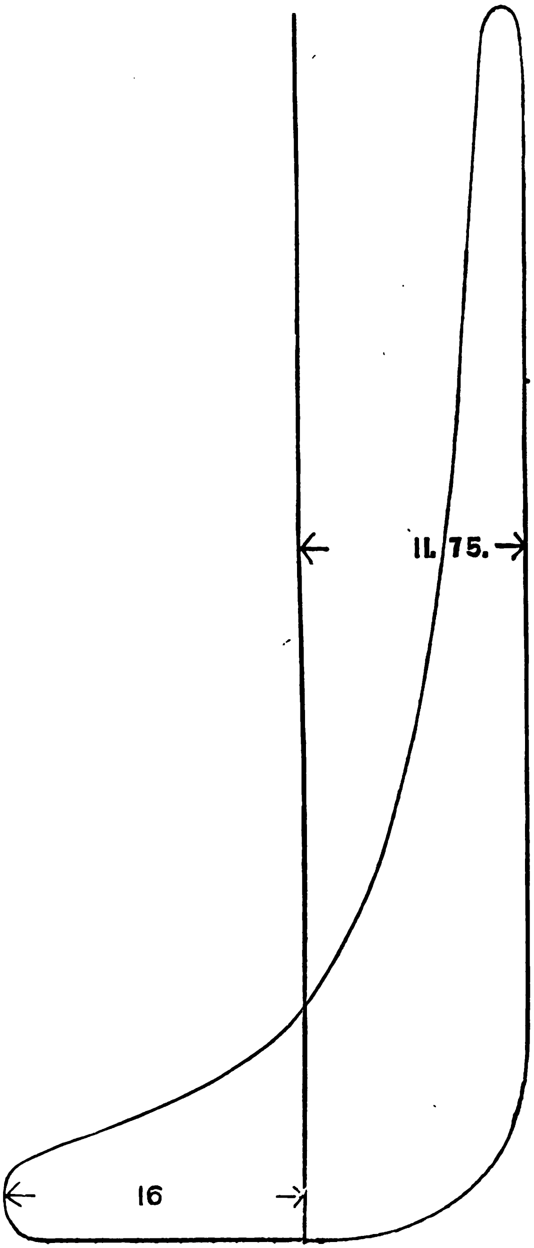


Diagram No. 22 is to the scale of 10lbs per inch, and Diagram No. 23 is to the scale of 12lbs per inch. It will be observed that the admission line in No. 22 is at right angles with the atmospheric

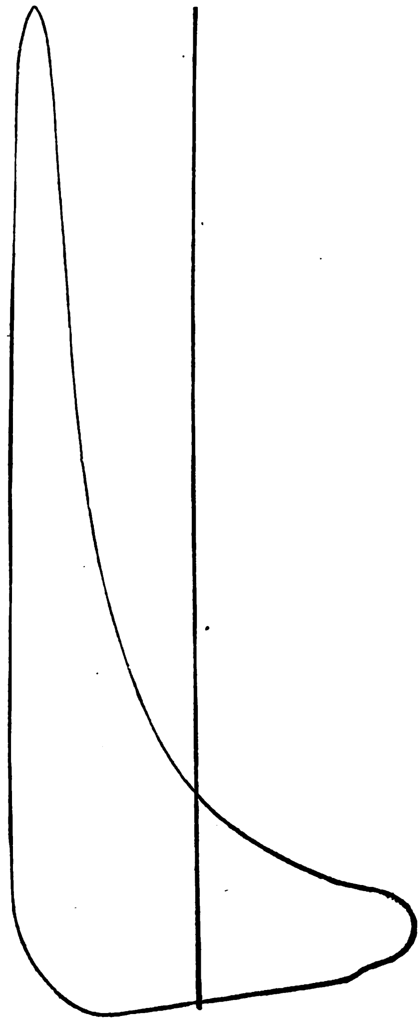
Diagram No. 22.



line to near the point of maximum pressure, where it gently rounds to the full initial pressure.

The admission line in No. 23 is seen to recede from the right angle line as though the admission of steam was too late. Indeed, if the

Diagram No. 23.



Indicator by which this diagram was taken had been implicitly relied upon, the tappet would have been moved forward in order to admit

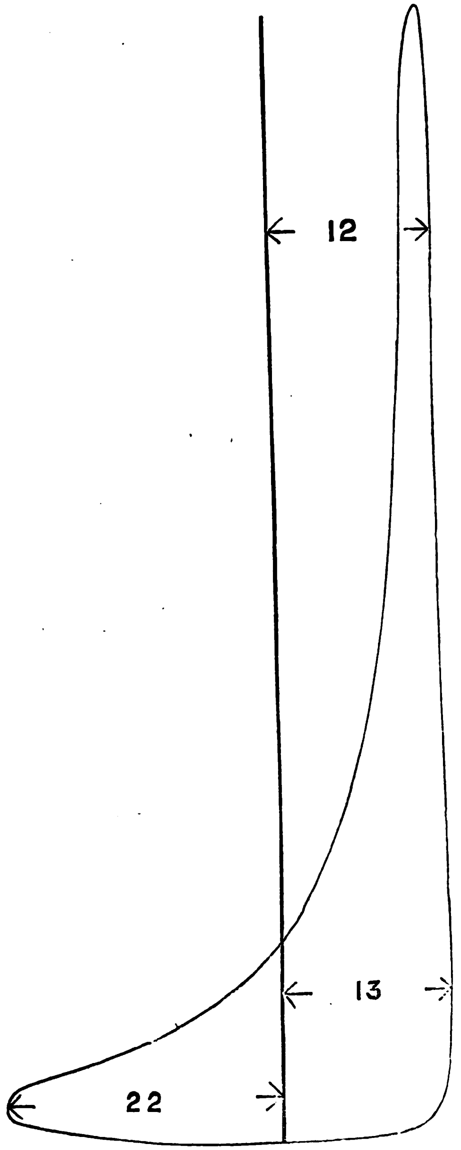
the steam earlier on the piston, the result of which might have been very disastrous, and would most certainly have caused a dangerous strain and vibration in the Engine. It is quite evident from Diagram No. 22 that such would have been the case, as the admission of the steam is here shown to be at the right time. If the piston of the Indicator were carried by its momentum, in advance of the actual pressure of steam, there would inevitably be a re-bound; and as this does not occur, there must be present the requisite sustaining pressure, which proves demonstrably that the valve is properly set, and not too late, as indicated by Diagram No. 23.

Diagrams No. 24 and 25 are from a Beam Engine in Rochdale, with Messrs. PETRIE'S Valves. No. 24 is an excellent example of valve setting, but the quality of vacuum is not good. It ought, with such an amount of expansion and terminal pressure, to maintain an average of 13lbs. No. 25 is from the opposite end of the cylinder, but as will be observed, is later in opening on the steam side and a little earlier in closing the exhaust. In this case an average of 10lbs of vacuum is obtained, and although the exhaust at this end is nearer the condenser, yet there is a positive loss of two pounds average pressure upon the piston throughout the whole of each exhaust stroke, at this end of the cylinder, being about one-sixth of the steam used; the cause, if not ascertained and rectified, will simply be a waste of steam to the extent shown. The value of an Indicator is thus again made manifest.

Diagram No. 26 is now shown in comparison to No. 24. The valve of this engine is of the construction known as the Back Slide Cut-off. There we have an example, not of better valve setting, as it is similar in all respects to No. 24, which was complained of in the former case of having only a 12lbs average vacuum, in this instance makes an average of 13lbs, with 27½lbs on the piston. In the case of No 24 the speed of piston is 329 feet per minute, and makes 29 revolutions per minute, whilst in this case the engine speed is 376 feet and 47 revolutions per minute. Now it is well known that it is more difficult to obtain an equal vacuum with increased revolutions of the engine, as that is more a consideration than the actual number of feet per minute. Thus we see that the one engine of 47 revolutions exceeds in every respect the other at 29 revolutions. For this discrepancy there is a cause and remedy; *although the initial* pressure is in excess of the other, yet the

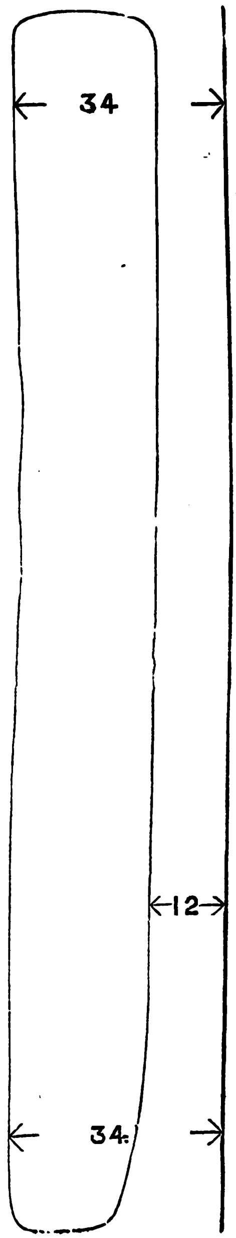
terminal pressure is about equal, therefore equal results should ensue.

Diagram No. 24.



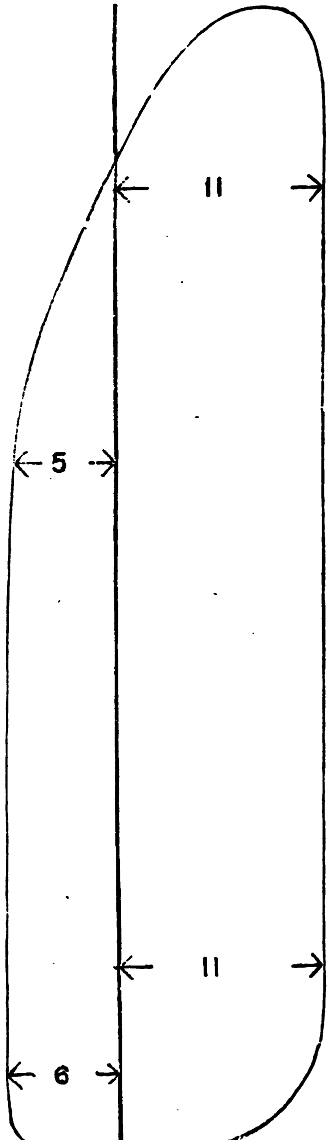
Diagrams 27 and 28 are from Compound Engines, and must there*fore be considered together.* The set of engines consists of

Diagram No. 27.



pressure cylinders we find the high pressure cylinders to be in capacity as 1 : 2·474 of the low pressure cylinders; the low

Diagram No. 28.

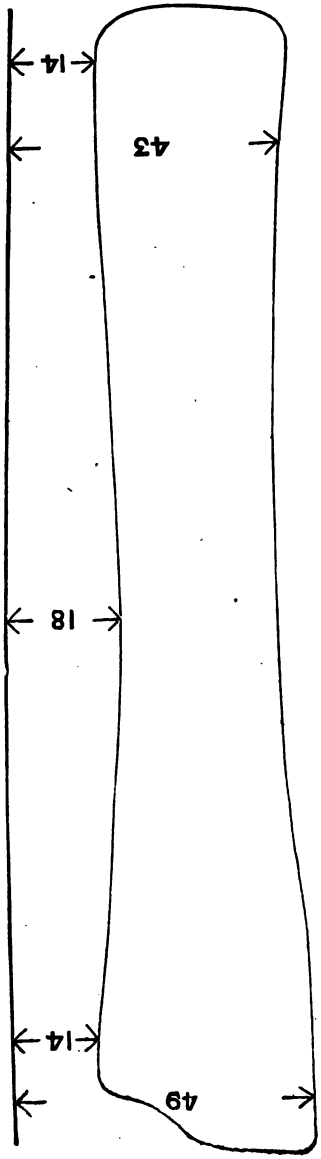


pressure cylinders being nearly two-and-a-half times the capacity of the high pressure cylinders. In these Engines nearly one-third of the steam is wasted. The steam is kept on at full pressure in the high pressure cylinder, during the whole length of the stroke; and in the low pressure cylinder, during two-thirds of it. This, as will be shown in another part of this work, is a very improper way of using the steam. The back pressure in the high pressure cylinder is more than is due to the initial pressure in the low pressure cylinder; and the amount of vacuum in the low pressure cylinder is not what it ought to be. The two sets of Engines are connected by gearing. The pipes which convey the steam from the high to the low pressure cylinders serve the purpose of a receiver, and thus maintain a nearly uniform back pressure.

Diagrams 29 and 30 are from the high and low pressure cylinders of a Compound Engine having the cranks placed at right angles, with the pistons beating at the same intervals of time; but, when one is at the end of its stroke, the other is in the middle. There is no receiver between the cylinders, the steam being conveyed direct from one cylinder to another by a pipe. The increased back pressure in the middle of the Diagram No. 29 is thus clearly accounted for. When the piston of the low pressure cylinder is at the end of its stroke—the high pressure piston being at the time in the middle, and the pipes between the cylinders being of small capacity—the pressure in the pipes, and also in the high pressure cylinder must rise, as there is no escape for the steam. The diagram from the low pressure cylinder, No. 30, has a strange peculiarity not seen in any of the preceding diagrams. From near the commencement of the stroke the pressure is seen to fall until it reaches about half the length of the diagram, from which point the pressure maintains a parallel line for another quarter of its length. This results from the same cause as the concave-curve back pressure line in the high pressure diagram. When the piston of the low pressure cylinder is in the middle of its stroke, and the high pressure piston at the end of its stroke, its exhaust valve is opened, and so an increased supply of steam is admitted to the low pressure cylinder at the middle of its stroke, which keeps up the pressure as shown in diagram. In some cases, with this arrangement of Engine, the pressure will actually rise again after having once fallen in the *low pressure cylinder*.—See Diagram No. 37.

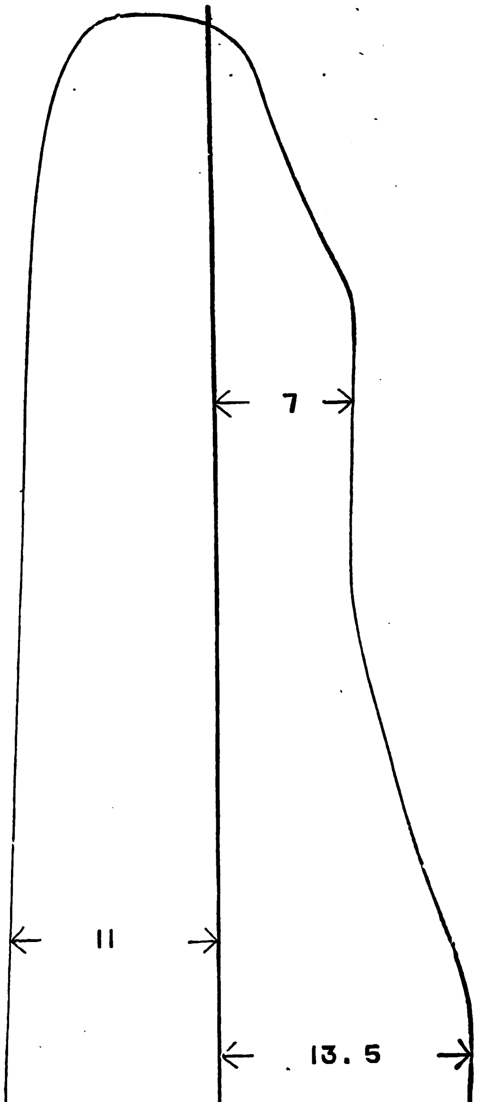
Diagrams 31 and 32 are from high and low pressure cylinders of a set of Compound Engines of the geared class. There is a pair of

Diagram No. 29.



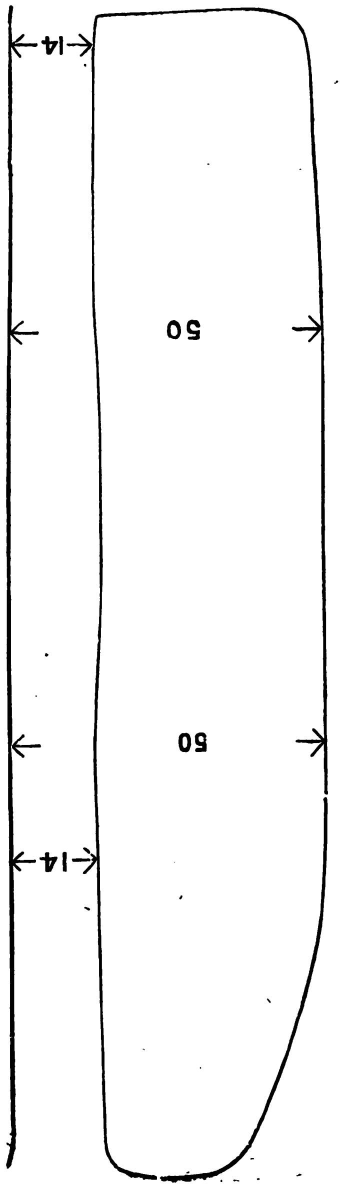
Beam Engines coupled together, with the cranks at right angles. In another room is a pair of Horizontal Engines coupled together,

Diagram No. 30.



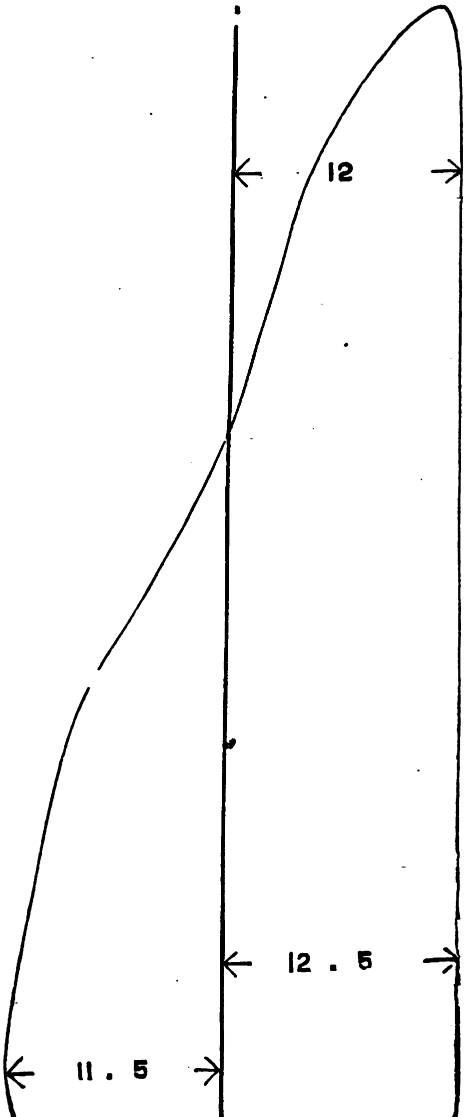
with their cranks at right angles, and running two and a half revolutions for one of the Beam Engines. These are the high pressure

Diagram No. 31.



cylinders, and their capacity, when the speed of piston is corrected to that of the Beam Engines, are as 1 to 3·4 of the latter. The two

Diagram No. 32.



pairs of Engines are coupled together by gearing. The pipes which convey the steam from the high pressure cylinders serve as receivers. The steam is cut off in the high pressure cylinder at about four-fifths the stroke. The back pressure, by reason of the capacity of the pipes, is nearly a constant quantity. There is a slight difference observable in the middle, when the back pressure rises about one pound. The cause of this is the increased speed of piston at that point. The speed is 70 revolutions per minute, and the ports are not large enough, or the valves are not open enough, to permit the steam to escape freely enough when the piston is at the highest speed.

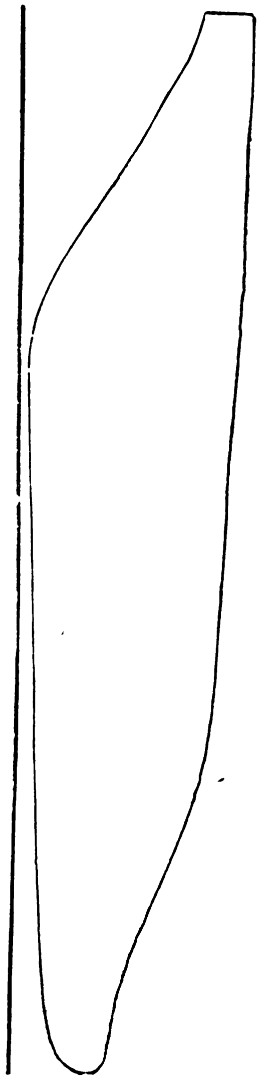
Diagram No. 32, from one of the low pressure cylinders, is not a bad one. It is much superior to many diagrams from Compound Engines, and very much better than its companion diagram, No. 30, from one of the high pressure cylinders. There is only a difference of 2½ lbs between the initial pressure in Diagram No. 31, and the back pressure in Diagram No. 30, which is as near as is usually got, but not as near as ought to be accomplished. These Engines admit of considerable improvement in connection with the high pressure cylinders.

Numbers 33 and 34 are from the top and bottom of a high pressure cylinder of a Compound Engine. They are given to illustrate a serious defect in the construction of the valves. An examination of the diagrams will clearly show this. At each end of the cylinder the exhaust valve remains closed until the piston has reached the end of its stroke. If it also closed too late, this defect could be remedied by an adjustment. This, however, is not the case, as the too early closing of the exhaust is still worse than the late opening. The exhaust valve should be open during half the revolution of the Engine, for, as it requires to open—say within one-tenth of the end of its stroke—so it should close when within one-tenth of the end of its return stroke. These diagrams show the exhaust valves to be open during about two-thirds of the stroke only, which has the effect of compressing the steam in the cylinder in a mischievous manner. This example proves the importance of testing Engines by the Indicator, even when fresh from the makers.

Diagrams 35 and 36 are from the high and low pressure cylinders of a MacNaughted Beam Engine. They may be classed amongst the best examples of the working of Compound Engines hitherto. The

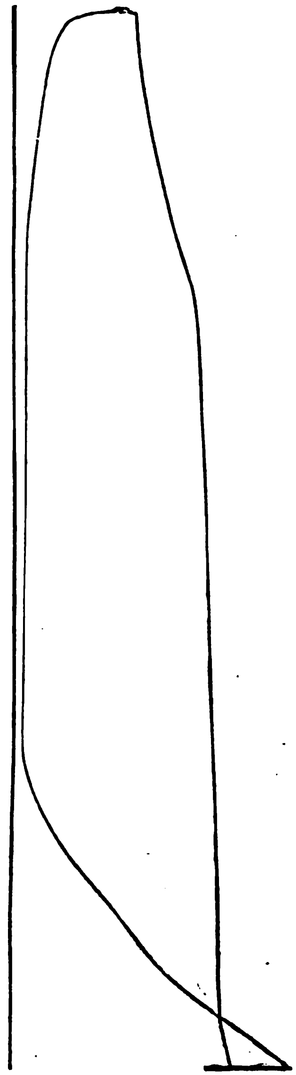
difference between the back pressure in the high, and the initial pressure in the low pressure cylinders, is very small, not being more than one pound. The diagrams are corresponding ones. The higl

Diagram No. 33.



pressure diagram is from the bottom, and the low pressure diagram from the top of the respective cylinders. The compression in the

Diagram No. 34.



high pressure cylinder is too great, and especially so when it is remembered that the speed is very slow.

Diagram No. 35.

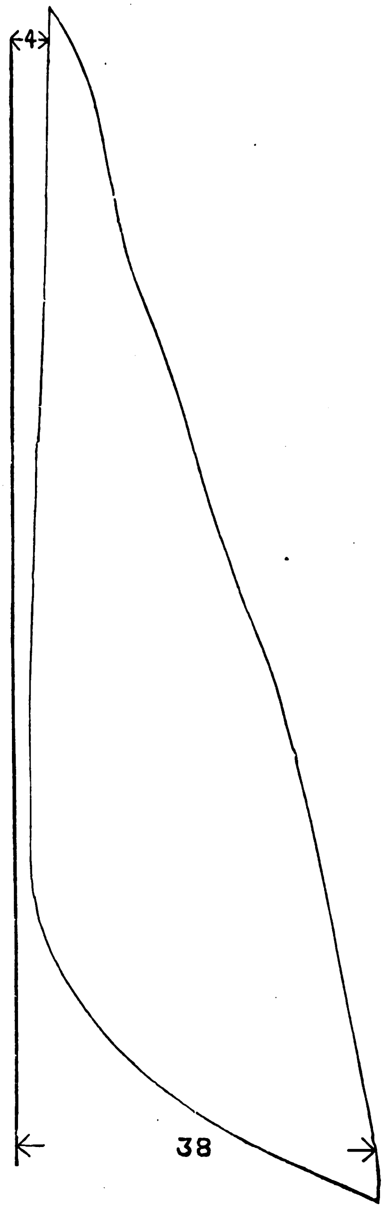
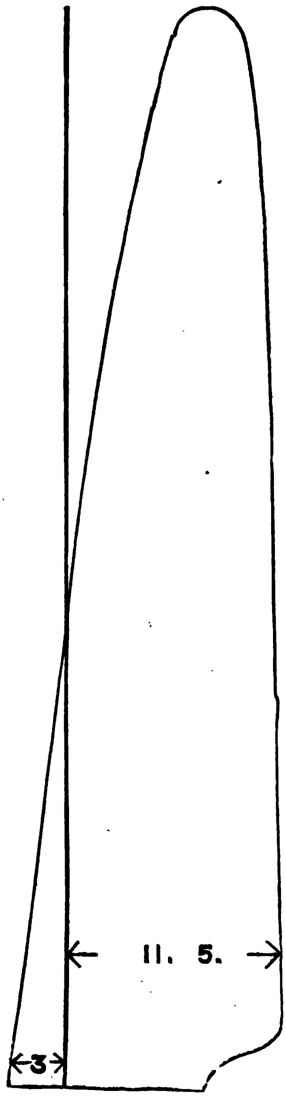


Diagram 37 possesses, but in a more striking degree, the characteristics of Diagram No. 29. It is from the low pressure cylinder of a Compound Engine, the pistons of which—though

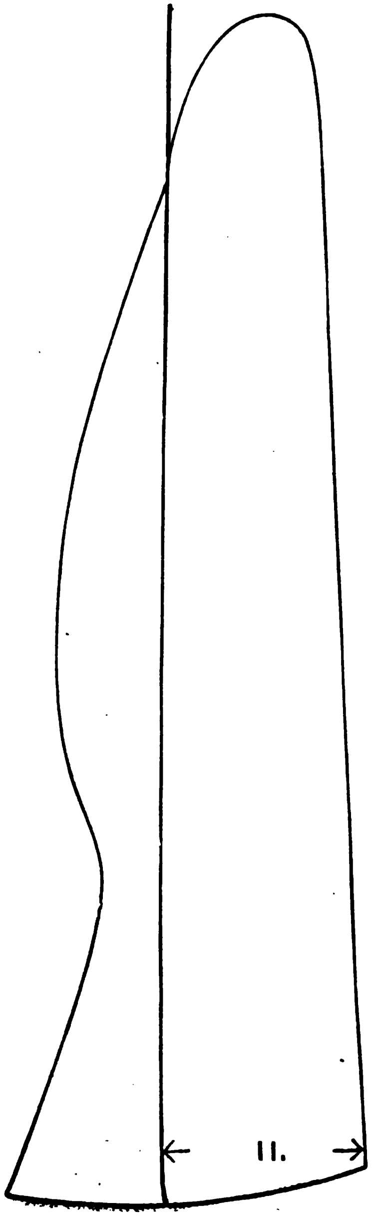
Diagram No. 36.



beating at the same intervals of time—do not beat simultaneously. In this case *the exhaust* valve of the high pressure cylinder

evidently opens some little time before the piston of the low pressure cylinder reaches the middle of its stroke, and before the latter has

Diagram No. 37.



attained its highest velocity; and, consequently, the fresh influx of steam raises the pressure again from the point to which it had fallen with its limited first supply. This is certainly not the best way of arranging a Compound Engine. To give an additional impetus to the piston at a time when it has attained its highest velocity is in utter disregard of the laws of dynamics.

Diagrams 38 and 39 are from a Compound Beam Engine. The high pressure cylinder is horizontal, and its crank is placed at the opposite end of the fly-wheel shaft to that of the Beam Engine. Diagram No. 38, from the high pressure cylinder, when viewed in connection with Diagram No. 39, from the low pressure cylinder, indicates very clearly that there is something wrong. It may be well to state at once that the corresponding diagrams, from the opposite ends of each cylinder, are exactly similar to these. Hence it is unnecessary to give all the four diagrams. The loop at the terminal point of Diagram No. 38 indicates leakage through the valves, or the piston, or both; and in the next place, the exhaust valve does not open soon enough, nor wide enough, to allow the steam to escape freely. This great back pressure cannot be due to the pressure of steam in the pipes which convey it from the high to the low pressure cylinder, because, if this should be the case, the greatest back pressure would be found at the termination of the exhaust line instead of the commencement of it, as we here find it. The initial pressure in Diagram No. 39, from the low pressure cylinder, when compared with the commencement of the back pressure line in No. 37, proves that a great loss of power is sustained.

Diagram 40 is a highly speculative one, and will usefully serve as a study to many Engineers. It is from the high pressure cylinder of a Compound Engine, which is vertical direct acting, with cylinders of 21 inches and 30 inches diameter respectively, and 3 feet 6 inches stroke, and running at a speed of 50 revolutions per minute. The capacities of the cylinders are in the ratio of 1:2. The valves of both cylinders are Cornish. The cranks are placed nearly opposite,—the high pressure crank leading by 18°, which equals one thirty-fifth of the length of the stroke, taking the mean of both ends. The high terminal pressure, and the still higher back pressure, are very singular phenomena, and merit some attention, as it is desirable to ascertain the cause. The apparent cut-off by the valves occurs *at about one-fourth* of the stroke, and at one-third it

Diagram No. 38.

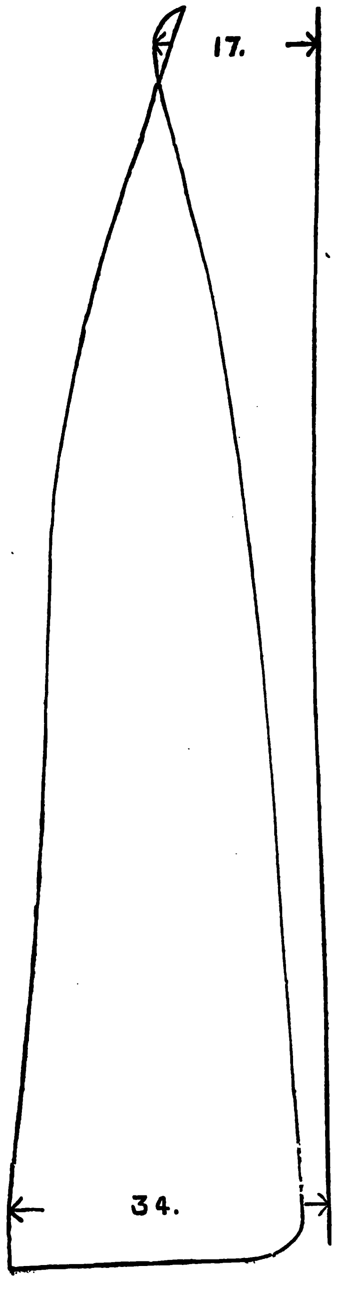
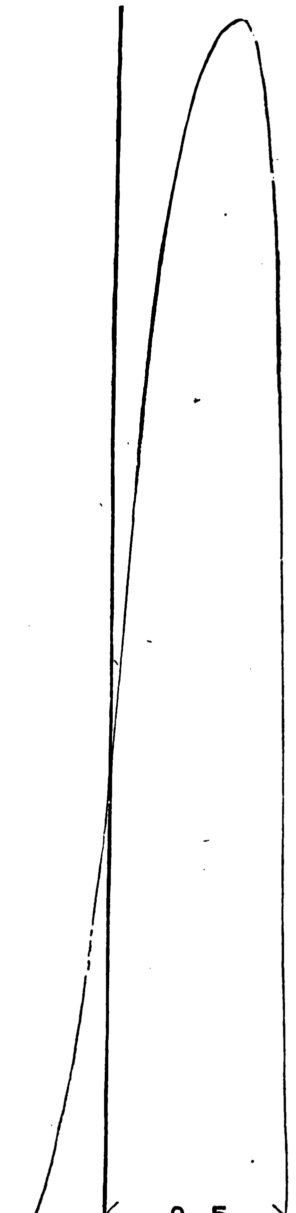


Diagram No. 39.



appears to show the cut-off complete. At this point the pressure is 38 + 15 = 53lbs absolute, which by the ordinary law of expansion

Diagram No. 40.

would give 17·66lbs terminal pressure, whereas the actual terminal pressure as per diagram is 18 + 15 = 33lbs. In order to make sure that the cut-off is really complete (judging and being guided by the form of diagram) let us take the middle of the stroke, where the pressure is found to be 23 + 15 = 38lbs. As a matter of fact the valves of the high pressure cylinder were made to cut off before half-stroke, and those of the low pressure cylinder to keep open the whole length of the stroke. Taking the pressure at the middle and dividing by two, we have 38 ÷ 2 = 19lbs, which should be the terminal pressure by the law of inverse ratio. This shows that a considerable quantity of steam is admitted between one-third and half-stroke.

The great increase of terminal pressure above what is due to the quantity of steam at half-stroke proves that steam is rapidly entering the cylinder during the time; and the rise in the pressure from 18 to 22lbs, whilst the piston is at rest at the end, is additional proof. It is quite evident from this that the valve is in a very defective condition, permitting steam to pass very copiously after it has been closed.

Why the pressure is not more rapidly reduced as the piston moves on its return stroke, is due to the same cause. As the valves of the low pressure cylinder remain open during the whole stroke, the pressure of steam at the termination of the stroke in the high pressure cylinder should be reduced on its back pressure or exhaust line in the proportion of the capacities of the cylinders at any point of the stroke. The ratios of the cylinders being as 1 : 2, it follows that when the low pressure piston has reached half-stroke, the steam, which at the commencement of the stroke of the high pressure piston occupied a capacity of one, will now occupy a capacity of 1·5, so that the pressure—which was 18 + 15 = 33lbs—should now be 33 ÷ 1·5 = 22lbs; or, taking the higher pressure at the commencement of the return stroke of 22 + 15 = 37lbs, and dividing as before, we have 37 ÷ 1·5 = 24·66lbs. We have here neglected the lead of the high pressure piston, because it would not affect the calculation appreciably. The back pressure line on the diagram at this point shows a pressure of 20 + 15 = 35lbs. This proves unmistakably either a great influx of steam, or the insufficiency of the valves to permit the steam to escape into the low pressure cylinder. Having a *full set of diagrams*, we are able to determine this point.

The low pressure diagrams show a mean initial absolute pressure of 31·5lbs, whilst the mean pressure in the small cylinder, at the same time, is 39·5lbs. The mean in the low pressure cylinder at half-stroke is 27lbs, and in the high pressure cylinder at the same time the mean pressure is 33lbs, which makes a difference of 6lbs, due probably to the friction of the steam in its passage through two sets of valves and the pipes—which are of small capacity, and lead direct from the one cylinder to the other without there being a receiver between them. If this loss of pressure in passing from one cylinder to another did not occur here, then the back pressure line would be reduced from 33lbs to 29lbs, which will be the equilibrium of the two pressures and capacities. From this data we shall be able to ascertain approximately the amount of leakage through the valves of the high pressure cylinder. The mean pressure of both diagrams at half-stroke on the advance line is 42lbs, at which point the valves have been closed as completely as their conditions permit. The terminal pressure, by the law of MARRIOTTE, will be 21lbs. When, however, we take the mean of the real terminal pressure—which we find to be 36·5lbs—and the pressure at half-stroke (42lbs), the quantity of steam which is found at the latter place is only 57·5 per cent. of that which is found at the end of the stroke, and before the pressure begins to rise whilst the piston is at rest. As 42·5 per cent. leaks through the valves during half the stroke when the pressure in the cylinder is highest, it is probable that such leakage is no less during the whole of the return stroke. This would lead to the conclusion that of all the steam which enters the high pressure cylinder, only 57·5 to ($42·5 \times 3 =$) 127·5, or 31,08 per cent. enters during the time which the valves are actually open,—indeed, up to half-stroke.

Let us see what is the proportion of leakage from another point in the diagrams. Above we have seen that the equilibrium of mean pressures, at the middle of the return stroke, is 29lbs. Now, at this point the steam occupies three times the capacity which existed in the high pressure cylinder at the middle of the advance stroke, so that following the law of inverse ratio, we should have 42lbs $\div\ 3 =$ 14lbs, whereas we find 29lbs to be the actual pressure. The calculated pressure of 14lbs, just arrived at, does not represent exactly the real pressure which would exist if no more steam entered *the high pressure* cylinder after the piston had passed the half-

stroke when advancing. The vario-thermal pressure line* would give 12·84lbs, and the actual pressure which would result from the condensation of some of the steam would reduce the pressure below this amount, so that if we say 12lbs, we shall probably be about correct. Then, as 12 is to 29, so is the quantity of steam which had entered the cylinder at one quarter of the revolution to that which has entered at three-quarters, so that we shall now find 12 : 29 :: 41·38 : 100. As we cannot take actual pressures at the end of the stroke in the low pressure cylinder, and as—for reasons which will be given—it would for this purpose be fallacious if we did, we will proceed to ascertain the total leakage from the last data. As at three-quarters of a revolution we have 29lbs, or 17lbs as the equivalent of the leakage for the second and third quarters of the revolution, then at the same rate, 17lbs, at the end of the fourth quarter, will become 25·5lbs; so that we get 12 + 25·5 = 37·5 as the final quantity of steam, instead of 12. The result will be as 12 : 37·5 :: 32 : 100. Thus, we arrive at the conclusion, that 68 per cent. of all the steam which passes into the cylinders is by leakage.

It will be useful now to bestow some little attention on the low pressure diagrams and cylinder. The most notable fact we discover here is the exceedingly small amount of vacuum—the mean average of the two diagrams (both ends) being only 6·5lbs. The cause of this we can only arrive at inferentially. The mean of the pressure just before the opening of the exhaust valve, for the escape of the steam into the condenser, being reduced to the terminal pressure, we find the latter to be thus, 18lbs. Referring once more to the high pressure diagrams in the middle of the advance stroke, where the pressure is 42lbs, and dividing by four, which will be the number of times to which it is expanded at the termination of the stroke in the low pressure cylinder, we get 10·5lbs as the final pressure. Correcting this to the vario-thermal pressure line, or pressure resulting from the increased volume of steam and its correspondingly reduced temperature, we get 9·41lbs as the final pressure; and, if we now make a small allowance for a still further reduction of pressure resulting from condensation in the cylinders, of say 0·41lbs, then the real final pressure would be 9lbs. This estimate is quite high enough for

* This is designated *by the late Professor* W. J. M. Rankine,—The Adiabatic Line, or Curve.

unjacketed cylinders—which these under consideration are—and as high as would be practically obtained. The result here arrived at gives us just half the amount of the actual terminal pressure in the low pressure cylinder, which the Indicator reveals, so that twice the amount of steam is present in the low pressure cylinder at the termination of the stroke as existed in the high pressure cylinder at half the forward stroke, and when the inlet valves had actually closed—so far as their condition permitted.

From this data we will now proceed to the consideration of the cause of the defective vacuum stated above.

By a preceding calculation we have conclusively shown that the steam entering the cylinder after the point of half-stroke, or termination of the first quarter of a revolution, constituted not less than 68 per cent. of the whole consumption—leaving 32 per cent. only as having entered at the proper time. The above examination of the real and calculated terminal pressures in the low pressure cylinder shows the proportion to be 50 per cent. As 32 per cent. would thus form only half, then 64 per cent. only of the steam which we have demonstrably proved to have entered the cylinders is found at the termination of the stroke of the low pressure cylinder. This will leave 36 per cent. as having disappeared. Where shall we look for it, or for an indication of the manner of its disappearance? The defective vacuum will sufficiently explain it. If the Engine had been in good and proper condition, the vacuum should have been not less than 12lbs; and as the season (being November) and other conditions were favourable, this amount, it is fair to assume, would have been obtained. As the vacuum shown by the diagrams is only 6·5lbs, then a deficiency of 5·5lbs has to be accounted for.

Now, as the valves of the high pressure cylinder are so very defective, we might reasonably suppose that those of the low cylinder would be so likewise. Coupling together the facts of the defective vacuum and the disappearance of 36 per cent. of the steam, it will be perfectly safe to conclude that this disappearance is due to the leakage of the valves which permit the steam to rush directly into the condenser on the exhaust side of the piston. This has the effect of wasting the steam by direct escapement, and also of neutralizing the force of that which would otherwise be effective on the steam side of the piston. The decrease of vacuum thus caused is no *guide to the amount* of steam consumed in causing it. The latter

will depend upon the amplitude of the exhaust ports and the efficiency of the condenser. The amount cannot be less than the decrease of vacuum, but it may be, and probably is, much more, as we will now show. The amount thus consumed, or rather lost, will be 10·125lbs certainly; this amount bearing the same proportion to 18lbs which 36 does to 64. If, now, we add 10·125lbs of steam to 5·5lbs of defective vacuum, the practical result will be an effective loss equal to 15·625lbs of steam. Then as 15·625 is to 18, so is the proportion of the non-effective consumption of steam to that which is effective—the loss of power being thus shown to amount to 46·75 per cent.

Since these last diagrams were taken which we have been analysing, the valves have been removed, and new ones of the grid form put in their places; and since going through the above analysis, we have obtained a set of diagrams very recently taken from the Engine in its changed condition, after it has worked for two years. We have also got full information as to the present condition of the Engine; the present and former consumption of coal, and amount of power. From these data we shall now be able to test the value and the correctness of the above analysis.

The average consumption of coal under the conditions which obtained when these diagrams were taken was 88 tons per month. The H.P. was not more than 130. After the change in the Engine, 5,000 mule spindles and preparations were added, raising the power to 180 H.P., and with a consumption of coal of 77 tons per month. The increase of power here amounts to 0·277 of the whole. The smaller consumption of coal is equal to 0·143 of the whole. Putting these two factors together, we get 0·420 as the proportion saved, which is simply 42 per cent. This is not quite so much as shown by the analysis and calculations above, and for a very good reason. The Engine, ever since the application to it of new valves, is not in a satisfactory state. The cranks are now placed at right angles, and there is not a receiver between the cylinders, the result of which is useless and excessive back pressure in the high pressure cylinder. The passage of the steam from one cylinder to the other is much impeded—the difference between the back pressure line in the first, and the steam line in the second cylinder, being nowhere (at corresponding points) less than 6lbs, and averaging much more. The want of true *proportion* between the two cylinders causes the

terminal pressure in the low pressure cylinder to be too high, so that the steam is not worked as expansively as desirable, and as economy requires. This also prevents as good a vacuum being secured as would otherwise be easily obtainable. The vacuum is now 11lbs average; and as the diagrams were taken in May, and the injection water drawn from a small lodge, which naturally has a much higher temperature than in November, (the month in which the former diagrams were taken) it will be seen that the assumption of 12lbs vacuum (potential) in a calculation above, was amply justified.

In view of all the facts of this case, we are fully warranted in stating, that with a correct proportion and size of cylinders, a still better arrangement of valves, and the addition of a receiver—more especially with cranks at right angles—the saving might be increased to 60 per cent. on the former consumption of fuel; that is, that 40 per cent. of the former consumption of fuel should have sufficed to produce the same power, by the simple alterations here indicated. Such enormous waste of steam as this case reveals could not be suffered to continue if the owners adopted the means to ascertain the facts. The Engine had been working in this state some years before the proprietors were aware of its deplorable condition; and as they had never possessed or used a Steam Engine Indicator, they were not conscious of the importance of indicating and understanding the diagrams.

In Compound Engines there is great necessity for indicating in order to secure a proper adjustment of the valves, otherwise much of the steam may be rendered non-effective. In many cases it will be desirable to have Indicator taps fixed on the pipes, which convey the steam from the high to the low pressure cylinder, in order to ascertain where the defects in the steam passages exist. The best places to fix them will be the valve-chests, or as near to them as circumstances will permit, so that they be outside the valves of each cylinder. This would enable the enquirer to see where the attenuation of the steam occurred. A difference of a considerable amount existing between the back pressure in the first, and the initial pressure in the second cylinder, it should be the aim of the Engineer first to discover the cause, and then to remove it.

We find many cases of unnecessarily great back pressure in the high pressure cylinder relatively to the initial pressure in the low *pressure cylinder of* Compound Engines. It should always be the

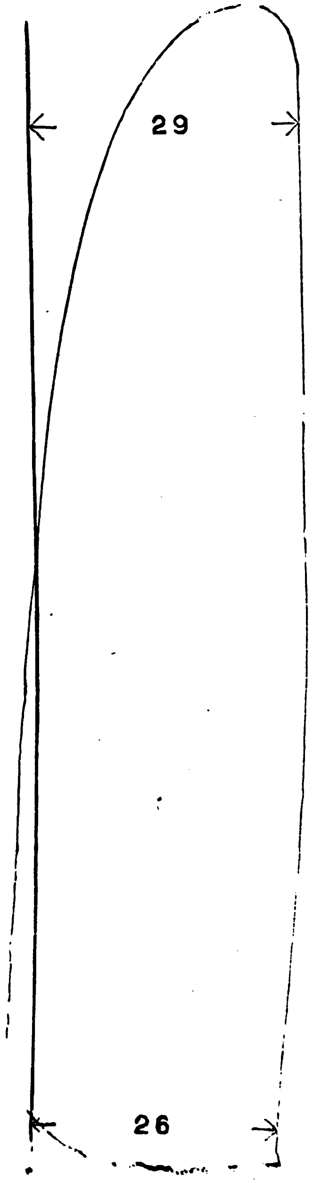
aim of the Engineer to have the back pressure line in the first, and the steam line in the second cylinder, to coincide as nearly as possible at the points which correspond in point of time, and up to the point of cut-off in the latter. It is very common to find great disparity in this respect. By attending to the recommendations herein contained, and comparing the diagrams thus taken with those taken in the ordinary way from the cylinders, the Engineer would be the better enabled to understand, and to rectify his Engine; whereas, otherwise he might be groping in the dark, and blundering in his efforts. The money thus saved to the owner, and the time, &c., saved to the Engineer, would no doubt be appreciated by both.

Number 41 is a common type of diagram from the high pressure cylinder of a certain species of Compound Engines (the most ancient of all the class of Compound Engines), the one originally patented by HORNBLOWER, and now usually known as WOOLF'S Plan. In this plan the cylinders are placed close together under the same end of the beam—the high pressure cylinder being nearest to the centre, and consequently having the shorter stroke. There is the strange anomaly in this diagram of the pressure rising as the piston advances for about one-third of the stroke,—and from this point a a constant pressure is maintained to the end. When the exhaust valve opens, the low pressure piston is also at the end of its stroke; and as the cylinders are close together, and have no receiver, there is no way for the steam to escape. Hence the great back pressure at the commencement of the exhaust line; giving a correspondingly high initial pressure in the low pressure cylinder, provided the valves of both cylinders be sufficiently open to permit the free passage of the steam.

Diagrams 42 and 43 are from the cylinders of a Horizontal Compound Engine belonging to Mr. ELI HEYWORTH, Audley Hall Mill, Blackburn, and made by Messrs. W. and J. YATES, Engineers, of the same town. The cylinders are placed parallel, and the cranks are fixed at each end of the fly-wheel shaft, the crank for the low pressure cylinder being 25 degrees in advance of the crank for the high pressure cylinder,—being equal to one twenty-fourth of the stroke (60 inches), or two-and-a-half inches traverse of the piston; so that when the high pressure piston is at the end, the low pressure piston has advanced two-and-a-half inches. The speed is 34 *revolutions per* minute, and the stroke of each piston

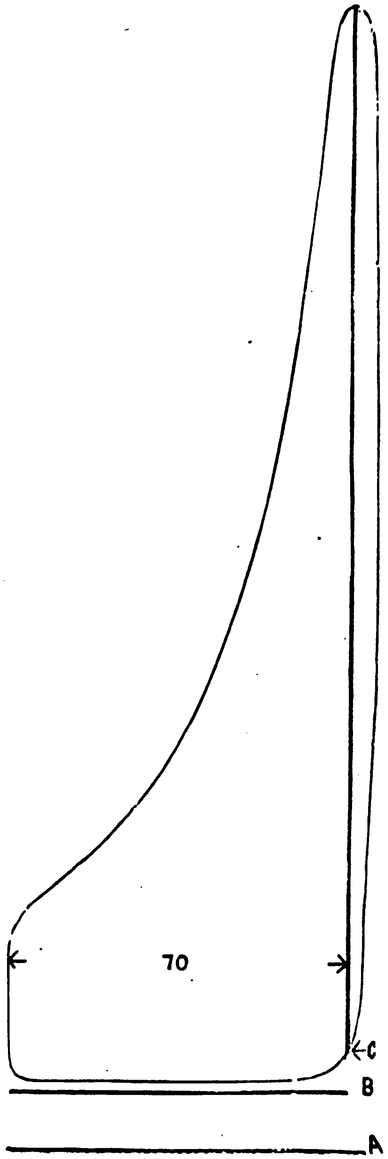
five feet, making 340 feet per minute speed of piston. The high pressure cylinder is 20 inches diameter = 314 inches area. The low pressure cylinder is 34 inches in diameter = 907·9 inches

Diagram No. 41.



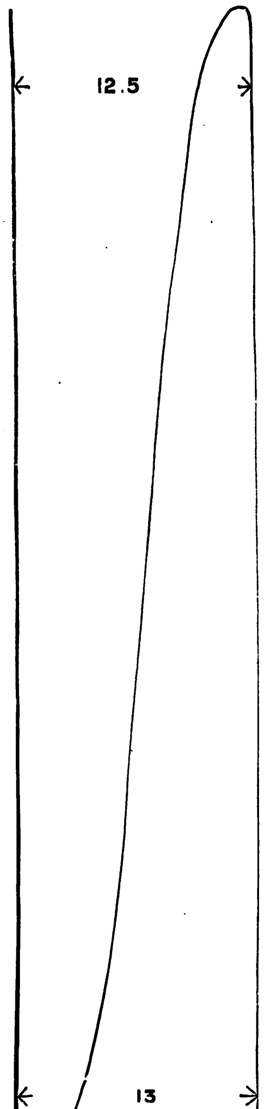
area. As the piston rods are each eight inches sectional area, and go through both covers, this must be deducted from the area of each cylinder. Then we shall have in even numbers—high pressure

Diagram No. 42.



cylinder 306 inches area; low pressure cylinder 900 inches area being in the proportion of 1 : 2·941. When the clearance, &c.,—tha

Diagram No. 43.



is, the whole fulcrum capacity—of each cylinder is taken into account, the relative capacities of the high and low pressure cylinders are found to be as 1:2·897. The clearance and thoroughfares at each end of the high pressure cylinder (being the whole fulcrum capacity) amount to 3·735 inches in length of its sectional area. This being added to the distance traversed by the piston at the point of cut-off, gives the measure of steam used at each stroke of the piston. The fulcrum capacity of the low pressure cylinder is equal to 2·785 inches of its sectional area.

The diagrams from each end are almost identically the same for both cylinders. The mean point of cut-off in the diagrams from both ends of the high pressure cylinder is found to be one 6·5 of the whole stroke, or 9·23 inches traverse of the piston; and adding 3·735 inches—the amount of the fulcrum capacity,—we find the measure of steam at the point of cut-off to be 12·965 inches; and as the length of stroke is 60 inches, add 3·735 inches = 63·735 ÷ 12·965 inches, which is the cut-off,—so far as the measure of steam—will be one 4·9 or 0·204 of the total steam capacity when the piston has reached the end of its stroke. The mean initial pressure of the front and back diagrams is found to be 70lbs above atmosphere, to which we must add 15lbs = 85lbs absolute initial pressure; and being divided by the point of cut-off, 4·9, will give the terminal pressure theoretically by the law of inverse ratio, at 17·34lbs, or 2·34lbs above the atmospheric line. Correcting this by the vario-thermal law, the terminal pressure would be 15lbs. The actual terminal pressure as shown by Indicator, is 18lbs. This shows an initial condensation of 16·67 per cent. of the steam which has entered the cylinder, and which is found at the end of the stroke in the state of steam. Whether any, and how much more, has been condensed and not re-vaporated, we cannot just now determine.

If the actual cut-off gave only the measure of steam shown by diagram, being one 6·5 of the stroke, the terminal pressure would then be 13·07lbs by the law of Marriotte, or nearly 2lbs below the atmospheric line. The theoretical average pressure when cut off at 0·204, and terminal pressure 17·34lbs, is 44·89lbs; and deducting 12lbs back pressure—which is about the average of the actual diagram,—the average effective pressure will be 32·89lbs. Now the Diagram No. 42, when measured by scale, gives an average pressure of *33·5lbs. If the* diagram, at the point of cut-off, gave the

whole initial measure of steam, the average pressure would then be higher than it is, because the fulcrum capacity which the steam must fill before the piston can be propelled by it, is not shown on the diagram. This space is shown by the faint line A, a little distance from the steam end of the diagram—and, if included in the measurement, would increase the average pressure from 33·5 to 39·375lbs. The theoretical diagram, with the cut-off at one 4·9 (0·204), the terminal pressure at 18lbs, and the back pressure 12lbs, would give an average of 34·6lbs. If the fulcrum capacity could be reduced to that which is required for clearance only, and which in this case amounts only to five-eighths of an inch, and shown in Diagram No. 42 by the faint line B,—then the average pressure would be 38·65lbs with the same measure of steam as shown by diagram, with fulcrum capacity added. This shows a loss of 13 per cent., which might be easily avoided by having the valves close to the cylinder. The loss here indicated is in the high pressnre cylinder only, because the steam is equally available for the low pressure cylinder as though the loss had not occurred. As the power at 33·5lbs average, with area of cylinder 306 inches, and 340 feet per minute of piston, amounts to 105·6 I.H.P., so 38·65lbs average will give 121·85 H.P.

Let us now turn our attention for a short time to Diagram No. 43, which is from the condensing, or low pressure cylinder. There is no receiver between the cylinders, hence it is necessary to keep open the valves of the low pressure cylinder for a considerable portion of the stroke. When these close, the back pressure in the high pressure cylinder begins to rise, as will be seen on reference to Diagram No. 42. It is necessary to understand this, otherwise the initial pressure in Diagram No. 43 might erroneously be supposed to be higher than the back pressure in No. 42. When the low pressure piston is at the end of its stroke, and its valve opening, the back pressure line in Diagram No. 42 is at the point marked with an arrow at C. At this point the exhaust valve has begun to close, so that the line which would correspond to the pressure in the pipes between the two cylinders, is disturbed. As the induction valve of the low-pressure cylinder is now closed, and the pipes are charged with steam, then the pressure at the point marked C would correspond almost exactly with the initial pressure in the low pressure cylinder, if the exhaust valve of the high pressure cylinder had not here begun to close.

The diagram here given, No. 43, is a correct representative of a considerable number from both ends of the cylinder, which, in their initial pressure, form, vacuum line, or back pressure, and average pressure, are almost identically the same. The cylinders—taking the whole steam capacity of each—are as 1 : 2·9 say; and as the terminal pressure of the high pressure cylinder is 18lbs, as shown by Diagram No. 42, the terminal pressure in the low pressure cylinder by the law of MARRIOTTE would be 6·2lbs. The actual terminal pressure would be 5lbs if the exhaust valve did not open before the end of the stroke. The theoretical terminal pressure, calculated by the law of MARRIOTTE, for the capacities of both cylinders, and the measure and pressure of steam used, would be 85lbs ÷ 14·21, which represents the number of volumes to which the initial volume of steam will expand, = 5·981lbs, or say 6lbs. Here then the actual terminal pressure is five-sixths of that which is given by the law of inverse ratio.

If we now carry the investigation a little further, we shall find a still more accurate and more surprising result. Turning to the table of pressures, temperatures, and volumes of steam, which have been established by REGNAULT, the celebrated French experimentalist, to whom we owe so much, we see at once the temperature and volume corresponding to any pressure under consideration. Now, as the initial pressure is 85lbs absolute, and as the number of volumes to which the steam is expanded is 14·21, as shown above—and as steam of 85lbs pressure has a volume of 340, so 340 × 14·21 =4,831, which will be the number of volumes occupied by the steam at the termination of the stroke in the low pressure cylinder. By the table, which will be found in appendix, this volume will be seen to correspond to a pressure of 4·75lbs per inch. By this vario-thermal law we assume that the temperature corresponds to the pressure and volume here found. This gives a very close approximation of the actual to the calculated terminal pressure, the difference being only five per cent., or in the ratio of 19 : 20. Taking the diagrams with initial pressure at 85lbs, and the final pressure at 5lbs, the ratio of expansion will be 85 ÷ 5 = 17.

It is important to state that these cylinders are steam jacketed around the cylindrical parts, but not on the covers. The casings thus provided are supplied with steam direct from the boiler, in which the *pressure is maintained* at 80lbs on the pressure gaug

and the steam which is condensed in the casings is returned direct to the boiler,—simply circulating. With these steam jackets some experiments have been tried, the results of which are highly interesting. One of the experiments was as follows:—The Engine was started after dinner *without* steam in the jackets. The steam in the boiler was maintained at a constant pressure. The cut-off valve in the high pressure cylinder was set at a given point, and continued there without change. The Engine was started at 1·30 p.m., and the steam was turned into the jackets at 1·40 p.m., and the speed of the Engine was counted at short intervals as under:—

At 1·40	the speed	was	33	revolutions	per minute.
,, 1·50	,,	,,	35½	,,	,,
,, 2·	,,	,,	37½	,,	,,
,, 2·15	,,	,,	37½	,,	,,
		STEAM	SHUT-OFF.		
At 2·30	,,	,,	36	,,	,,
,, 2·50	,,	,,	35	,,	,,
,, 3·0	,,	,,	34	,,	,,
,, 3·20	,,	,,	33½	,,	,,
,, 3·40	,,	,,	33	,,	,,

The governors were disconnected during the time, and the throttle valve fixed wide open. The amount of machinery was kept constant and unvarying, so that the increased speed of the Engine was due solely to the effect of the steam in the jackets of the cylinders. The experiment undoubtedly proves a gain in the power of the Engine; but whether the gain be due to the greater efficiency of a given quantity of steam, or to a larger consumption of steam in a given time, is not apparent at once, and therefore requires a full and close investigation, which will be made a little further on, when a few more facts have been given, in the light of which this question will be more satisfactorily determined.

The amount of steam condensed in the jackets has also been ascertained. The condensed water was carefully collected for one hour, and found to be 4¼lbs weight. This is 4¼lbs reduced from steam at a temperature of 322° to water at 212°. Now it can easily be estimated what would be the amount of coal required to generate steam at 322° from water of 212°. We find that 4¼lbs of water in steam at 70 + 15 = 85lbs will give 39,882 cubic inches; and as *each stroke, according* to calculations already given, consumes 3,978

inches, it follows that 4¼lbs will be equivalent to 10 strokes, or five revolutions of the Engine. We have seen that the Engine makes 4½ revolutions per minute more with steam in the jackets. Now 4½ revolutions per minute will give 270 revolutions per hour; and as the steam required to produce this is the equivalent of five revolutions, then we have 270 to 5, or a gain of 54 at the cost of one. This is undoubtedly true economy if all the data be correct, and great care has been taken to secure the correctness of them. Still, the small weight of water here given as the result of condensation in the jackets creates a strong suspicion that this is not the truth,—but that possibly—nay probably—some undetected escapement occurred. Whatever may be the truth in this matter, the other tests to which the Engine has been subjected will not be affected by it.

Diagrams Nos. 42 and 43, which we have discussed at some length, were taken with steam in the jackets. We have taken a considerable number of diagrams *without* steam in the jackets. The steam was shut-off at night, and not admitted next day until the diagrams had been taken. The cut-off valve had been left as working with steam in, and when the Engine was counted the speed was found to be slow, and the cut-off valve had to be altered to cut-off later in order to get the same speed as when Diagrams Nos. 42 and 43 were taken. These diagrams thus taken are an interesting study, and show curious results. They give a total of about five I.H.P. more than with steam in the jackets, although exactly the same machinery was at work. In one case 161 I.H.P., and in the other case 166 I.H.P. The low pressure diagrams are the same in their initial and terminal pressures as when taken with steam in the jackets. The diagrams from the high pressure cylinder are about 5lbs less in their initial pressure, and 2lbs more in their terminal pressure, and give a slightly higher average pressure.

This clearly proves, in two ways, greater initial condensation. First, with steam in the boiler at the same pressure, the initial pressure in the high pressure cylinder is lower. The higher terminal pressure shows a greater quantity of steam admitted,—the steam which was condensed at the commencement of the stroke having been more or less re-evaporated at the end of the stroke. The low pressure cylinder cannot do this to the same extent without external aid, as the difference in *temperature* between the initial pressure and

the exhaust line, is only half as much in the low as in the high pressure cylinder. The temperature of the steam in the high pressure cylinder ranges from 316° to 188°, being a difference of 128°; and in the low pressure cylinder from 188° to 126°, being a difference of 62°. It is here assumed that the temperature corresponds to the pressure for saturated steam as given in the tables of REGNAULT. As the amount of condensation is assumed to be as the difference of temperature of the steam at initial and back pressure lines; and as the ratio of expansion is greater in the high than in the low pressure cylinder, it must be allowed that the greatest proportion of condensation will occur in the high pressure cylinder for a given area of surface, as compared with the low pressure cylinder,—other conditions being the same.

There is one more noteworthy feature in these diagrams which merits attention. It has been already stated that *without steam* in the jackets of the cylinders, the terminal pressure in the high pressure cylinder was 20lbs; whereas, when steam was *in* the jackets, the terminal pressure was 18lbs. Now, the noteworthy fact is this, —that the diagrams from the low pressure cylinder are the same under both conditions, the average being 6·2lbs, and the terminal pressure 5lbs absolute. The obvious conclusion to which these two facts lead us is, that when the casings of the cylinders are not supplied with steam, then a larger proportion of the steam which passes from the high to the low pressure cylinder disappears in the latter without performing work. This is a study of great interest and importance. Whatever may be the conclusions arrived at by theoretical calculations concerning the isothermal, vario-thermal, or thermodynamic curves, the Indicator supplies the only means for ascertaining the actual curve or pressure of steam during the course of its expansion.

As these diagrams represent one of the best cases of economical working to be found in the country on the compound (or any other) principle, this somewhat lengthy discussion of them the reader will not deem superfluous; and it may be still further useful if we now compare this case with the result which would be obtained by compounding according to the law and rule of proportion given in chapter on compounding. For the purpose of this comparison we will take Diagrams 42 and 43 as our guide, so far as the arrangement of cut-off; and also the dimensions and speed which here

obtain. The calculations in both cases we will make on a theoretical basis, so that the comparison may be absolutely exact and perfectly trustworthy. We will assume the same initial pressure of steam in the cylinder; the same cubic capacity of steam for each stroke; and the same number of expansions. The low pressure cylinder being the standard of power, as explained in the chapter on compounding, must be the same area in each case. Taking the diagram, No. 42, in which the cut-off is at one 6·5 or 0·15384 of the stroke, and adding to it the clearance of $\frac{6}{8}$ths of an inch, which is necessary, the point of cut-off—so far as the real initial capacity,—will then be at one 6·2, and this we will fix upon as the basis of the calculation. The cylinder, as will be remembered, is 306 inches sectional area (available), and 60 inches traverse of piston. The cylinders being in the ratio of 1 : 2·9, then $6·2 \times 2·9 = 17·98$, which is the ratio of expansion, that is, the number of volumes to which the steam will be expanded.

First, let us Calculate the theoretical power of a compound Engine, with the proportions of cylinders, &c., as above. The initial pressure being 85lbs $\div$ 6·2 = 13·71lbs terminal pressure. As the capacities of the cylinders are as 1 : 2·9; and as we find the steam in the low pressure cylinder to be cut off at 0·666 or two-thirds the stroke; and as two-thirds of 2·9 will be 1·93; so the steam in the high pressure cylinder, with a terminal pressure of 13·71lbs, passing into the low pressure cylinder, which, to the point of cut-off has a capacity 1·93 times greater, the initial pressure in this cylinder will be 7·1lbs, and the terminal pressure 4·73lbs. This will give 17·98, or, say, 18 expansions. Now the high pressure cylinder, with a terminal pressure of 13·71lb, and cut-off at one 6·2 of the stroke, will give an average absolute pressure of 38·717lbs; but as the initial pressure in the low pressure cylinder is 7·1lbs, so this will be the back pressure, which must be deducted from 38·717lbs, leaving 31·617lbs as the average effective pressure on the piston. The low pressure cylinder, with a terminal pressure of 4·73lbs, and the cut-off at two-thirds of the stroke, will give an average of 6·645lbs, from which we deduct 2lbs back pressure (making it equal to a vacuum of 13lbs), giving an average effective pressure of 4·645lbs. The result can now be expressed by the equations,

HIGH PRESSURE CYLINDER.

$$\frac{306 \text{ inches} \times 31{\cdot}617\text{lbs} \times 340 \text{ feet}}{33000} = 99{\cdot}68 \text{ H.P.}$$

LOW PRESSURE CYLINDER.

$$\frac{900 \text{ inches} \times 4{\cdot}645 \times 340 \text{ feet}}{33000} = 44{\cdot}338 \text{ H.P.}$$

Together making $= 144{\cdot}018$ H.P.

The measurement of steam used to the point of cut-off in the high pressure cylinder was one 6·2 (or 0·1613) of 60 inches = 9·677 inches × 306 in area = 2,961 cubic inches.

Let us now see what will be the power given out by the same initial pressure of steam, the same cubic capacity, and the same ratio of expansion, and therefore the same terminal pressure,—arranged according to the rule of proportion for Compound Engines given in chapter on compounding. The low pressure being taken as the standard for the power required, the cylinder will be 900 inches area as before. As the steam is expanded 18 times, the high pressure cylinder will require to be 210 inches area, and the point of cut-off will be one 4·25 or at 14·11 inches. This will give 2,961 cubic inches measurement as before. Then 85lbs initial pressure, cut off at one 4·25, will give a terminal pressure of 20lbs, and an average absolute pressure of 48·94lbs. The low pressure cylinder being 4·25 times the area of the high pressure cylinder, and the steam being cut off at one 4·25, the initial pressure will coincide with the terminal pressure in the high pressure cylinder, and will also be the amount of back pressure there. The initial pressure in the low pressure cylinder being 20lbs, and cut off at one 4·25, the average pressure will be 11·5lbs. The average absolute pressure of the high pressure cylinder being 48·94lbs − 20lbs = 28·94lbs, which will be the average effective pressure on the piston. The low pressure cylinder will give an average absolute pressure of 11·549lbs, from which must be deductive 2lbs for back pressure, or defective vacuum, leaving 9·549lbs as the average effective pressure on the piston. The result will now be expressed by the equations,

HIGH PRESSURE CYLINDER.

$$\frac{210 \text{ inches area} \times 28{\cdot}94\text{lbs} \times 340 \text{ feet}}{33000} = 62{\cdot}615 \text{ H.P.}$$

LOW PRESSURE CYLINDER.

$$\frac{900 \text{ inches} \times 9{\cdot}549\text{lbs} \times 340 \text{ feet}}{33000} = 88{\cdot}545 \text{ H.P.}$$

Together = 151·160 H.P.

This arrangement here shows a better result than the former one by 4·73 per cent. The Compound Engine from which Diagrams 42 and 43 are taken, proves by its arrangement of cut-off, &c., to be a very near approach to the true law of expansion, giving, as we now see, 95·27 per cent. of the result of the true proportion. If made according to the rule of proportion (see chapter on compounding), it would be still more economical, and the power would be more usefully and scientifically apportioned between the two cylinders. At present, as we have seen, the high pressure cylinder is giving 69 per cent. of the total power obtained. Now, by arranging according to the law of proportion, as given in rule, we not only obtain the greatest duty from the steam, but we get the power correctly apportioned between the two cylinders. In the case here last calculated, the high pressure cylinder gives 62·615 H.P. and the low pressure cylinder 88·545 H.P. being for the high pressure cylinder 41·42 per cent., and for the low pressure cylinder 58·58 per cent. of the total power. When we remember that the reciprocating parts of the low pressure cylinders are always, and almost necesasrily, of greater weight than those of the high pressure cylinder, it is evident from a consideration of the dynamic laws discussed in the chapter on compounding, that these latter proportions are certainly the best.

A very conclusive test has been applied in connection with the steam jackets of the Engine under consideration, which it is not necessary to do more than refer to here, as the facts are fully stated in the chapter on steam jacketing, to which the reader is advised to turn for the details. Suffice here to say, that the test is the increase of temperature given to the injection water, and that the saving is 13·3 per cent. with steam in the jackets.

One part of the case in connection with these diagrams, stated some little way back, requires further analysis. The question is that of increase of speed of the Engine when steam is supplied to the jackets of the cylinders. The assumed gain by increased speed when steam is in the jackets, is hypothetical, as the increased speed would presumably require a proportionately increased quantity of steam, and certainly would do, if each stroke consumed the same

quantity as before. This, however, cannot be the case, as a larger measure, and still larger real quantity, is used at every stroke for the same speed when steam is not in the jackets;—the difference in the quantity, as shown by the terminal pressure in the high pressure cylinder, being as 10 : 9, that is, the quantity of steam used at a given speed *with* steam in the jackets is 90 per cent. of that used when steam is *not* in the jackets. The difference of 10 per cent. here found at the termination of the stroke of the high pressure cylinder will not represent the full difference, for the following reasons. When the cylinder is kept hot by the steam in the jackets, half the internal area at the point of cut-off is thus economically affected, because the face of the piston and the cylinder covers are the same under both conditions. Obviously the initial condensation will be greater when the walls of the cylinder are not maintained at a high temperature by steam being in the jackets; and equally obvious is it that the re-vaporation of the condensed steam will not be so great. This view is still further supported by the fact, that when steam is *not* in the jackets, the lower pressure cylinder only produces the same initial and average pressure as when—being heated,—it receives a smaller quantity of steam. The same conditions obtaining, the same law will operate in each cylinder. Therefore it may be fairly concluded that a larger quantity of steam has entered the cylinder in this case than in the former, relatively to the terminal pressure. How much more cannot here be determined.

The increase of speed from 33 to 37½ revolutions per minute, represents an increase from 87 to 100. This corresponds very closely with the conclusions drawn from the terminal pressure in the high pressure cylinder, when allowance is made as required by the considerations just presented; and the correspondence with the result of the injection water test is exact and complete.

The gain assumed by increase of speed seems doubtful at first sight, because, the cylinder being at each stroke of equal capacity to the point of cut-off (the cut-off remaining unchanged), it would appear that an equal quantity of steam of a given pressure in the valve chest would be taken in at every stroke, and therefore that an increase of speed would simply represent a proportionately increased consumption of steam. We have here to consider a disturbing cause, which has an important bearing on the question before us, and

which sensibly affects the result. This disturbing cause is the steam in the jackets. When the steam is admitted into the cylinder which is *not* thus heated, a larger proportion of condensation—both initial and permanent—takes place than when the cylinder *is* heated. This will be clearly seen on a full consideration of the conditions. As the steam in expanding and producing work loses a portion of its heat thereby, it is necessarily condensed, and the water which is now in suspension in the steam—being a good conductor of heat—carries away with it a considerable amount of heat from the metal of the cylinder. When a fresh supply of steam now enters the cylinder it is largely condensed by contact with the cooled surfaces of the metal, and the rapidity with which heat is transmitted from the steam to the metal is in proportion to the pressure,—which is the measure of the activity of the atoms. The amount of initial condensation will then depend upon the quantity of heat to be supplied to restore the equilibrium, if the time which elapses be sufficient to permit this. Now, it is quite clear that, if the internal surfaces of the cylinder can be maintained at a sufficiently high temperature by external means, then a smaller amount of initial condensation will certainly occur.

From these considerations we shall be the better led to a true appreciation and comprehension of the question of increased speed of the Engine when steam is supplied to the jackets. If we suppose that the valve orifice is just sufficient to admit the requisite quantity of steam for maintaining the speed of the Engine when the cylinders are externally heated, than when this is not done, and an excess of initial condensation results, a less amount of steam will be available for power, and a slackening of the speed will be the consequence. By this more time is given for the ingress of steam, and therefore it is quite reasonable to suppose that as much steam is actually consumed for a given time as when running at the higher speed.

Nothing has yet been said about the economy of this Engine. Fortunately we are permitted to give the facts of the case, which every reader will be happy to learn. The consumption of coal (the ordinary Lancashire Engine coal) is in summer 13·25 tons, and in winter 15·5 tons per week. If we say 14·5 tons average for the whole year, we shall not be overstating the case. The I.H.P. we have already given at 160 — 165. It has been given in the reports of one of the *Boiler Insurance* Associations at 180 I.H.P. The

establishment consists of a shed containing 740 looms, and other buildings necessary for cotton manufacturing. The taping is done here, not only for these looms, but for some which are in operation in another mill,—making altogether 1,000 looms. Now the additional 2½ tons of coal consumed during the winter half of the year is evidently required for keeping up the temperature of the rooms in excess of that which is required during the other half of the year. As the taping is usually allowed to require one ton of coal for every hundred looms, we will allow in this case half a ton, which will give us, for this purpose, five tons per week. We will estimate the consumption of coal for keeping the rooms warm, and maintaining the boiler fires, when the Engine is not running, at 2½ tons per week average for the whole year. These two items together make 7½ tons per week. The total consumption being 14·5 — 7·5 = 7 tons per week required for power. Then 7 tons = 15,680lbs ÷ 60, the number of hours which the Engine, runs per week, = 261·33lbs coal per hour, ÷ 160 I.H.P., = 1·633lbs coal per H.P. per hour. The value of the result here given depends upon the correctness of the estimates for deductions, which those in the trade can estimate for themselves. But even if we appropriate for power 9 tons, leaving only 5½ tons for all other purposes to (which is inadequate), the consumption only amounts to 2lbs per I.H.P. per hour.

The co-efficient of expansion in these diagrams is worthy of notice, being an example of efficiency and economy. The high pressure cylinder has an average of 33·5lbs. As the ratios of cylinders are as 1 : 2·9, so 33·5 ÷ 2·9 = 11·55lbs, which, being added to the pressure in the second cylinder, will give 6·2 + 11·55 = 17·75lbs as the effective average pressure for the low pressure cylinder only. As the absolute terminal pressure is 5lbs, therefore 17·75 ÷ 5 = 3·55, which is the co-efficient of expansion. Correcting this by the rule for finding the co-efficient of expansion given in chapter on compounding, we shall get a slightly different result. The cut-off, as per Diagram 42, being at 1 — 6·5, and the ratios of cylinders as 1 : 2·9, then 2·9 × 6·5 = 1·85 expansions. Initial absolute pressure 85 ÷ 18·85 = 4·509lbs. The actual terminal pressure being 5lbs, the mean of the two figures will be 4·754lbs. Then 17·75lbs average effective pressure ÷ 4·754 = 3·733, as co-efficient of expansion.

In connection with these diagrams we have now advanced facts and speculations sufficient, we believe, to deserve, and ensure, a careful study.

Diagram No. 44 was taken on the 19th February, 1873, from one of the Side-Lever Engines (being the low pressure cylinders of Compound Engines) belonging to Messrs. GEO. CHEETHAM and SONS, Staley Bridge. It is given as an example of a good vacuum. Though the Engineer (Mr. ENOCH GLEDHILL) secures at all times the best vacuum which his experience enables him to attain, yet this is the best one in his collection. Full details are here given to enable the reader to understand it better.

Barometer .... =	30·46 inches.
Vacuum Gauge =	29·6 ,,
Diagram 14·1℔s =	28·728 ,,
Hot Well	69° temperature.
Injection Water..	37° ,,
Difference	32° ,,

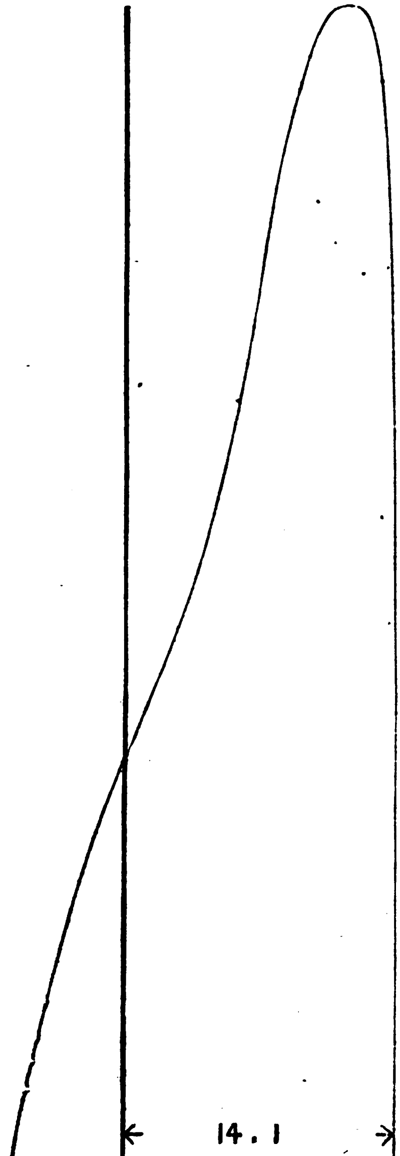
The diagram is taken to the scale of 10lbs per inch.

Diagrams Nos. 45 and 46 are from one of the Engines belonging to the world-known establishment of Sir TITUS SALT, BART., SONS, and Co., Saltaire, near Bradford. They are Beam Engines, of the simple (not compound) principle, in pairs of two cylinders each. The cylinders are each 50 inches diameter, 7 feet stroke, with a speed of 30 revolutions per minute, making a speed of piston of 420 feet per minute. The valves are the Corliss, with INGLISS and SPENCER'S patented improvements, and made by Messrs. HICK, HARGREAVES, and Co., Bolton. The cylinders are steam jacketed, both around the cylindrical part and at the top and bottom,—so that they are entirely surrounded with steam casing, and are charged with steam at the boiler pressure.

The cut-off in No. 45 is at one-tenth the traverse of the piston, as per diagram. The initial pressure of steam is 28lbs + 15 = 43lbs absolute. At the first division of ten, where the steam is completely cut off, the pressure is 26lbs + 15 = 41lbs absolute, and this we will take as the initial pressure, on which to base our calculations in the following analysis. The actual terminal pressure, as found by the diagram, is 6lbs absolute, or 9lbs below the atmospheric line.

Now, *the point of cut-off* being one-tenth, and the length of the

Diagram No. 44.



stroke being 7 feet, = 84 inches, then $84 \div 10 = 8{\cdot}4$ inches traverse of the piston at the point of cut-off. As, however, the clearance, &c. (the whole of fulcrum capacity), is equal to one inch sectional area of the cylinder, it follows that the space filled with

Diagram No. 45.

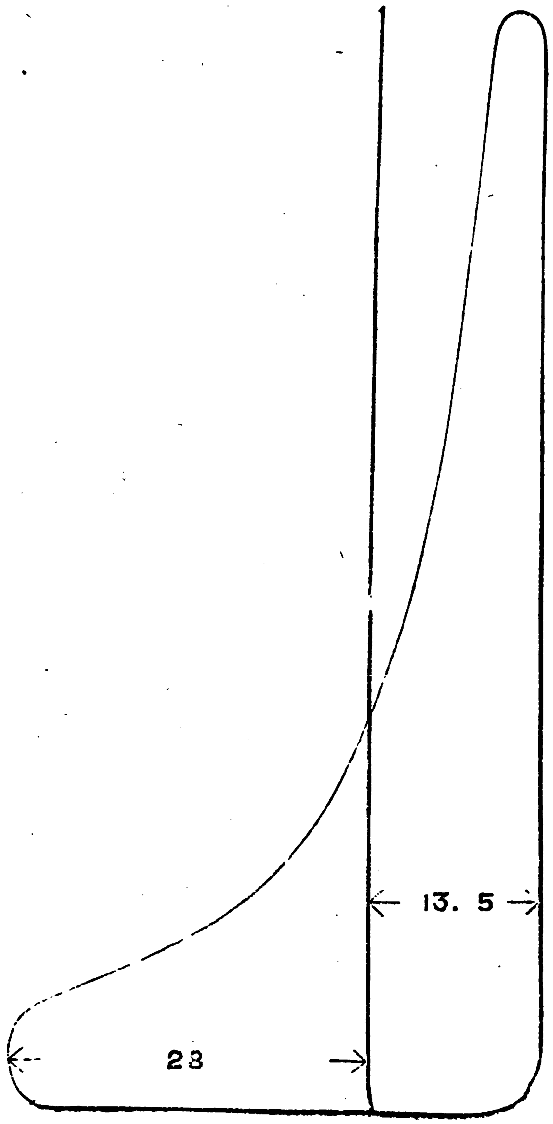
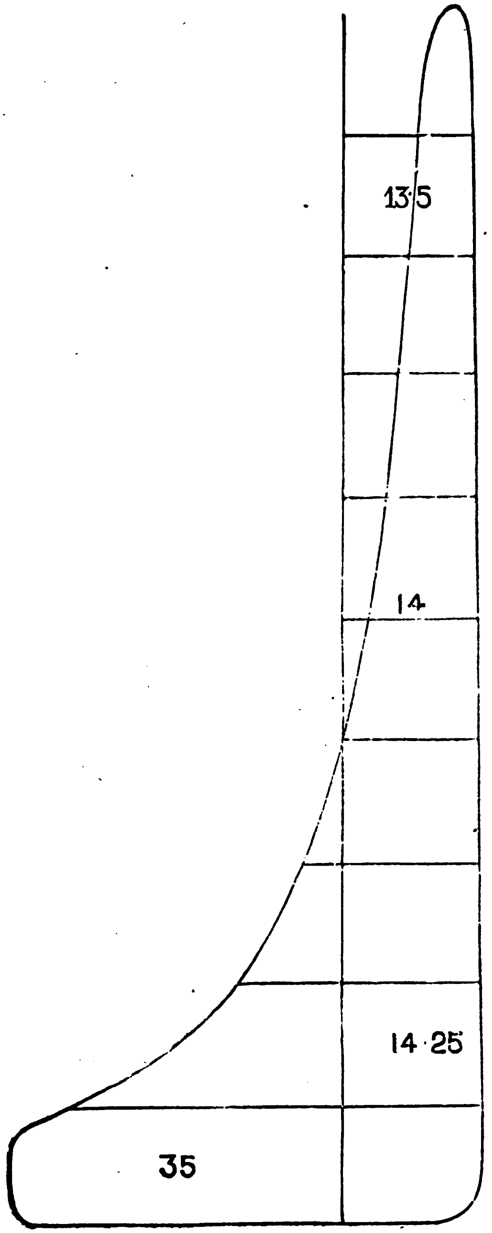


Diagram No. 46.



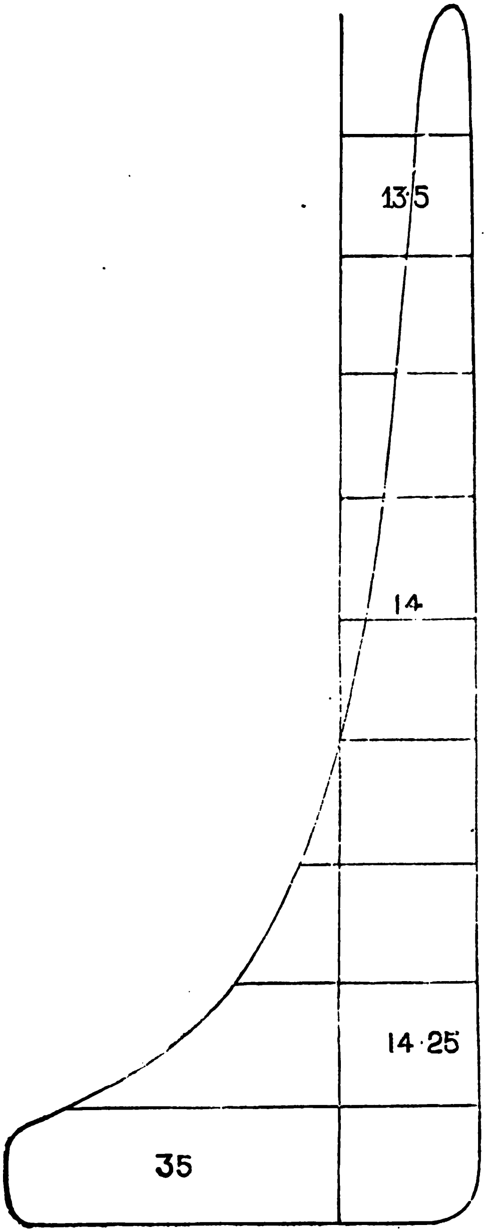
steam when the cut-off occurs will be 8·4 + 1 = 9·4 inches. The stroke being 84 inches + 1 inch = 85 inches, which will be the length of cylinder filled with steam when the piston has reached the end of its stroke. Taking the length as just found, and dividing by the length at the point of cut-off, we shall have 85 ÷ 9·4 = 9·04, which will be the real point of cut-off so far as the measure of steam, and which also will be the number of volumes to which the steam will expand.

The absolute initial pressure, as already shown, is 41lbs, which, being divided by 9·04 will give 4·537lbs as the terminal pressure by the law of MARRIOTTE, or inverse ratio of volume to pressure. This, however, notwithstanding that the cylinders are steam jacketed, will not be the terminal pressure practically obtained. The vario-thermal pressure line is what would obtain if no other disturbing cause operated, and for the following reason.

If we suppose that no condensation takes place on the ingress of the steam, and before the point of cut-off (this is only *supposed* for the purpose of illustration), yet condensation must result as the piston advances, and as the steam expands, by reason of the work performed. This is a law that applies to all steam which produces work whilst expanding. As the steam is condensed it will at once be more or less re-evaporated, because of the heat contained in the metal of the cylinder as the effect of steam jacketing. Steam, in the presence of water from which it is generated, has always a temperature and pressure corresponding. By reference to the table in appendix it will at once be seen that the higher pressure of steam, and consequently higher temperature, has a volume greater than that which is given by the law of inverse ratio. That the steam at the terminal pressure has a temperature corresponding to such pressure is beyond doubt, because, as condensation must take place as the piston advances, it is clear that the presence of watery particles in the steam would necessarily produce such temperature. Furthermore, there may be, and probably is, more or less condensed steam in the state of water, carried in suspension in the steam; and this would absolutely prevent a higher terminal temperature than corresponds to the terminal pressure. Such being the case, it may be accepted as a law (provisionally) that the temperature will be as the pressure at the end of the stroke, for every degree of expansion.

Correcting the calculated terminal pressure, as given above by

Diagram No. 46.



steam when the cut-off occurs will be 8·4 + 1 = 9·4 inches. The stroke being 84 inches + 1 inch = 85 inches, which will be the length of cylinder filled with steam when the piston has reached the end of its stroke. Taking the length as just found, and dividing by the length at the point of cut-off, we shall have 85 ÷ 9·4 = 9·04, which will be the real point of cut-off so far as the measure of steam, and which also will be the number of volumes to which the steam will expand.

The absolute initial pressure, as already shown, is 41lbs, which, being divided by 9·04 will give 4·537lbs as the terminal pressure by the law of MARRIOTTE, or inverse ratio of volume to pressure. This, however, notwithstanding that the cylinders are steam jacketed, will not be the terminal pressure practically obtained. The vario-thermal pressure line is what would obtain if no other disturbing cause operated, and for the following reason.

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Correcting *the calculated* terminal pressure, as given above by

the law of MARRIOTTE, of 4·537lbs, to the vario-thermal pressure line required by this latter law—and exhibited in table of pressures, temperatures and volumes,—we shall now get a terminal pressure of 3·8lbs per inch, and this would be the real terminal pressure if the steam at the point of cut-off was pure, without admixture of water, and if all that which is condensed during the remaining part of the stroke should be re-evaporated at the termination. As the actual terminal pressure is 6lbs, it is evident that the steam which is found at the point of cut-off, and which occupies a sectional area of cylinder equal to 9·4 inches, represents only 63·4 per cent. of the steam (or steam and water) which has really entered the cylinder at the point of cut-off; or, to state it differently, of the total amount of steam which has entered the cylinder at the point of cut-off, 36·6 per cent. of it has been condensed, and the whole of this has been re-evaporated at the termination of the stroke.

This proportion would be represented by 5·44 inches sectional area of the cylinder, in addition to the 9·4 inches of actual capacity; and if represented graphically on the Indicator diagram, a line at right angles with the atmospheric line, and parallel with the admission line, must be drawn at a distance beyond the latter of 0·766 of the distance from the admission line to the point of cut-off, or a little more than three-quarters of a division (one-tenth) of the diagram.

It is more than probable that all the steam condensed is not re-evaporated, as will be shown hereafter, and if so, the condensation will be proportionately greater than has yet been indicated.

This diagram (No. 45) is one of the most beautiful of its kind, and shows a perfection of valve arrangement and valve setting, and general efficiency which it would be difficult indeed to equal, and impossible to exceed. The vacuum is almost the best attainable, and proves the admirable condition of the Engine. Two diagrams from the same Engine, which we have lately received from GEORGE SALT, ESQ., show a vacuum of 14·25lbs,—the best we have ever seen.

These latter diagrams reveal a fact which greatly strengthens the conclusions drawn above, with regard to the condensation of the steam on its admission. The boiler pressure is higher, and consequently the initial pressure in the cylinder is higher, being 35+15=54lbs absolute, and therefore the cut-off is earlier, as

we should naturally expect, when there is only the same amount of power to be produced. Limiting our investigation to one of the two diagrams (No. 46), the cut-off is shown to be at one-thirteenth of the stroke. As the stroke is 7 feet, = 84 inches, then 84 ÷ 13 = 6·46 inches, which is the distance traversed by the piston to the point of cut-off. he whole fulcrum capacity (clearance, &c.) being equal to 1 inch sectional area of the cylinder, then 84 + 1 = 85 inches, which is the length of cylinder filled with steam at the termination of the stroke. The traverse of the piston to the point of cut-off being 6·46 inches, + 1 inch sectional area for fulcrum capacity, will equal 7·46 inches filled with steam at the point of cut-off. Then, 85 inches ÷ 7·46 = 11·4, which will be the real point of cut-off as to the measure of steam, and the number of volumes to which the initial volume will be expanded.

As the initial pressure is 50lbs, and the number of expansions 11·4, then 50lbs ÷ 11·4 = 4·386lbs, which would be the terminal pressure by the law of MARRIOTTE, or inverse ratio. If we now correct this terminal pressure to the vario-thermal pressure line, as in Diagram No. 45, and by the same process of calculation, we shall find that the real terminal pressure would then be 3·6lbs. which is very nearly what was found for Diagram No. 45.

Now the actual terminal pressure by the diagram (No. 46), under consideration, is 6·75lbs. This will give a relatively larger initial condensation than in the former case. As 3·6 is to 6·75, so is the amount of steam represented by the length of initial capacity, 7·46 inches, to the amount which actually enters the cylinder up to the point of cut-off. As 3·6 : 6·75 :: 53·33 : 100.

From this we see that the volume of steam in the cylinder at the point of cut-off represents only 53·33 per cent. of that which has really entered, and which is found at the termination of the stroke. This being so, then it follows that 46·67 per cent. of the steam which enters the cylinder up to the point of cut-off has been condensed, and is then in the state of water in suspension in the steam.

In the case of Diagram No. 45, where the real (not apparent) cut-off was found to be at 0·1106—which is the reciprocal of 9·04, the number of expansions,—the initial condensation is 36·67 per cent.; whilst in this case, with a higher initial pressure and an earlier cut-off, being at 0·0877, the reciprocal of 11·4, the condensation *amounts to 46·67* per cent. Here are results of a curious and

suggestive character. It should be remembered that all the conditions of the cylinder, with respect to steam jacketing, &c., are the same in the two cases, except one,—this one condition being the ratio of expansion of the steam. It is certainly extraordinary that the proportion of initial condensation which takes place is almost exactly in the ratio of the degree of expansion. Whether such proportionate condensation would obtain for every degree of expansion is altogether problematical. A somewhat singular feature in these diagrams is worth a passing notice. In Diagram No. 45 the calculated terminal pressure by the vario-thermal law, is 3·8lbs per inch. The actual terminal pressure as per diagram is 6lbs. In the second diagram, with an earlier cut-off, the calculated terminal pressure is 3·6lbs, and the actual 6·75lbs, which shows a greater proportionate re-evaporation.

We are not at all sure that the full amount of initial condensation is by this analysis detected, because, so far as the data which have been given, we only discover so much of the condensation as comes to be re-evaporated, and is found at the end of the stroke in the state of steam, and revealed by the Indicator. When the cylinders are not supplied with steam in the jackets, as is sometimes tried experimentally, then a smaller proportion of condensation is detected by the diagram, and yet a much larger amount of condensation has really occurred, as is proved by other data. Diagrams taken without steam in the jackets, and having the same initial pressure, have the cut-off later, and therefore have a larger initial volume of steam, and yet the terminal pressure is just about the same. This being so, then, either there is a less amount of initial condensation, or, there is less re-evaporation. That the latter is the true explanation does not admit of a single doubt. If, when the ends of the cylinders are maintained at a high temperature by a supply of steam direct from the boilers, the steam is copiously condensed on its entrance into the cylinders, then it is certain that the condensation will be still greater when the metal is cooler. This is so self-evident that no one will venture to call it in question. If any lingering doubt should remain, we have irresistible evidence to prove that such is indeed the fact.

A great number of very exact experiments have been made on these Engines, by George Salt, Esq., to whose courtesy we are *indebted for being* enabled to present a variety of valuable evidence

in connection with them. One notable series of experiments, conducted over a sufficiently extended period to ensure trustworthiness, was to determine the difference in the economy of the Engines *with* and *without* steam in the jackets. First, the water from the condensation of steam in the jackets of one pair of Engines (two cylinders) was carefully collected and weighed, during one whole week of 59 working hours. This water amounted to three tons per day of 10·5 hours; and to 16·85 tons for the week. During the same period the pair of Engines were supplied with steam from boilers which were disconnected from the rest, and used only for this purpose. The coal was carefully weighed, and the feed water was as carefully measured. The consumption of coal for the week was 72 tons, and the weight of water evaporated was 1,122,688lbs, or 501·2 tons. In this case, the proportion condensed in the jackets, to the whole steam consumed, is equal to 0·0336, or a little more than one-thirtieth. This is the proportion of condensation ascertained by the data before us. It follows, that if with any other arrangement a greater quantity of steam passes through the cylinders for the same amount of power, and with the same terminal pressure on the diagram, then, clearly, such excess of steam has disappeared in the cylinders by condensation.

During another entire week, the Engines and boilers were carefully tested *without* steam in the jackets, the weight on the Engines being maintained exactly as before, even to the numbers being spun. The consumption of coal was 84 tons, and the water evaporated proportionately greater. The evaporative efficiency of the boilers in both cases being 6·96lbs water per 1 lb of coal. The proportion of coal and steam consumed in these cases is as 7 in the latter to 6 in the former case. The weight of water here amounts to 584·7 tons per week, all of which passes through the (two) cylinders. In the former case, 501·2 − 16·85 = 484·35 tons per week only passed through the cylinders. This leads to the inevitable conclusion, that without steam in the jackets, not only is there a greater aggregate consumption of steam in the ratio of 7 to 6, but also, that (584·7 − 484·35 =) 100·35 tons, being 0·1712, or a little more than one-sixth of the whole quantity consumed, is permanently condensed in the cylinders, and passes out in the state of water—not being r evaporated. The ratio of condensation (ascertained) in the for case (16·85 tons in the jackets) is 0·0336, or a little more than *thirtieth of the whole* consumption.

The supposition may arise in the minds of some readers that the high terminal pressure, which is found relatively to the initial pressure and volume may be the result of leakage through the valves. This condition is so frequently found to exist that the idea is worthy of consideration. As the diagrams at both ends of the cylinders are similar in all respects, then if any leakage should occur, it may be presumed to be about equally distributed amongst all the induction valves. Diagram No. 45 has a vacuum of 13·5lbs; and if the atmosphere at the time of taking it should have been a 14·7lbs, which is the average atmospheric pressure, then 14·7 − 13·5 = 1·2lbs,—the difference between the pressure in the cylinder and a perfect vacuum.

When it is remembered that an eminent Engineering authority lately declared before a learned and practical body of men—the Institution of Mechanical Engineers,—that "the steam could not be transferred from the cylinder to the condenser, with less than a difference of 2lbs in pressure" (of course this is absurdly incorrect); and, the difference between the cylinder and the absolute vacuum will be still greater;—remembering this, the supposition of leakage, if it has been entertained, will seem entirely baseless. But when we contemplate the more recent diagram, in which the vacuum is 14·25lbs, even allowing the pressure of the atmosphere at the time to have been 15lbs—leaving only a difference of 0·75lbs,—then the supposition is still more peremptorily dismissed. It is important to say, that to ensure the correctness of this last diagram, every precaution and care had been taken,—such as the exact testing of the Indicator Springs. One interesting, and we may venture to say, unique, fact, deserves to be recorded here. The same Indicator after taking the diagram, was immediately disconnected from the cylinder, and attached to the pipe leading direct from the condenser, when the pencil moved on the very same line, except at the commencement of the exhaust, where the deviation was only just perceptible.

It will be interesting now to ascertain by the actual terminal pressure of steam, what is the weight of water in the state of steam, which passes through the cylinders during one week. For the determination of this we will take the second diagram referred to, the terminal pressure of which is 6·75lbs, because it was taken under the conditions of the amount of power, steam in jackets, con-

sumption of coal, and evaporation of water, given above. As this diagram is a fair representative of the set, we may make the calculations from it for the pair of Engines.

The volume of steam at 6·75lbs is 3,500 to 1 of water at its greatest density. The cylinder capacity will be as follows:—Stroke 84 inches, + 1 inch fulcrum capacity = 85 inches in length, diameter 50 inches = 1,963 inches area. Allowing 26 inches sectional area for the piston rod, and dividing by two, as it is only found on one side of the piston, we get 13 inches, which being deducted from the area above, 1,963 − 13 = 1,950 in area of cylinder available. Then 1,950 inches × 85 inches = 165,750 ÷ 1,728 = 95·92 cubic feet. As the Engines run at the speed of 30 revolutions per minute, then each cylinder will give 95·92 × 60 = 5,755·2 cubic feet capacity per minute. Then 5,755·2 × 60 = 345,312 cubic feet per hour; and multiplying this by two we get 690,624, which is the cubic measurement of both cylinders in one hour. Dividing this by the relative volume of steam to water, we shall have 690,624 ÷ 3,500 = 197·321 cubic feet of water. Now a cubic foot of water at its greatest density (39·1° fahrenheit) weighs 62·4lbs; therefore, as 197·321 × 62·4 = 12,312·83lbs, this will be the weight of water per hour, which leaves the two cylinders in the state of steam. The weight of water per week will thus be 12,312·83 × 59 hours = 726,457lbs of water per week. To this amount must be added that which was ascertained to have been condensed in the jackets, 16·85 tons = 37,744lbs + 726,457 = 764·201lbs.

The water evaporated by the boilers which supply only these cylinders being, as shown above, 1,122,668lbs per week, there is manifestly a deficiency of 358,487lbs, which amounts to 32 per cent. of the total evaporation of water. This is the proportion missing of the total evaporation; but in order to ascertain the proportion which disappears in the cylinders it will be necessary to present the factors above in a different order. As the total evaporation is 1,122,688lbs; and as the amount passing through the jackets is 37,744lbs; then 1,122,618 — 37,744 = 1,084,944lbs will be the weight of water which passes through the cylinders. Deducting from this the factor found above, we shall now have 1,084,944 — 726,457 = 358,487lbs, which is the weight of water carried out of the cylinders in suspension in the steam—constituting 33 per cent. of all that enters. The weight of steam (*that is* in the gaseous state) passing out of the

cylinders, as shown by the diagrams, constitutes 64·7 per cent. of total amount evaporated. Where has this large proportion of steam and water gone? Assuming the data to be correct (and every care and precaution have been exercised to ensure this), then the inevitable conclusion presents itself, that the steam has been abundantly liquefied in the cylinders, and that, notwithstanding the steam jacketing and consequent large re-evaporation, yet this large proportion—32 per cent.—is in the state of water in suspension in the steam immediately before leaving the cylinders. From this last conclusion proceeds another.

As, without steam in the jackets, there is a larger consumption of steam in the ratio of 7 : 6, the weight of water evaporated per week (of 59 working hours only) being 1,309,803lbs; and as the whole of this must pass through the cylinders; and as the same pressure of steam only is found at the termination of the stroke; then 1,309,803 — 726,457 = 583,346lbs, which will be the weight of water in suspension in the steam just before it passes into the condenser, and which in this case constitutes 44·54 per cent. of the total evaporation; or, the steam detected by the Indicator is only 55·46 per cent. of all which has been generated, and which has passed into the cylinders. The result of the analysis just made may now be usefully presented thus: Of the whole of the water which is evaporated in the boilers when the Engines are working under the two different arrangements, the proportions found at the end of the stroke are,

With Steam in the Jackets		64·7	per cent.
Without do.	do.	55·46	,,

The difference, amounting to 14·3 per cent., or in the ratio of 7 : 6.

If we were now able to ascertain the actual weight of water and steam which passes into the condensers, we could then see the value of the above calculations. As they are surface condensers, and as all the steam is condensed and kept separate from the water which serves for the purpose of condensing, then this datum might possibly be obtained.

There is one feature in these diagrams not yet noticed, but which we shall treat in the chapter on compounding; that is, the co-efficient of expansion, or the proportion which the absolute *terminal* pressure bears to the average effective pressure.

...turn to the question of initial condensation in the ... different data have been given by which to ... quantity of steam. First, the initial volume in the ..., the terminal volume; third, the weight of water ..., and consequently, weight of steam generated. By the ...ond data we ascertained that the initial condensation ... the weight of steam found in the second, was—

With Steam in the Jackets 36·67 per cent.
Without do. do. 46·67 do.

...e quantity found by the second datum, relatively to the whole ...nt which entered the cylinders, was—

With Steam in the Jackets 67·00 per cent.
Without do. do. 55·46 do.

The final conclusion with regard to the initial condensation can now be presented. By taking the ratio which the first datum bears to the second, and the second to the third, we find that the total initial condensation (which is the proportion of water held in suspension in the steam at the point of cut-off) will be—

With Steam in the Jackets 57·56 per cent.
Without do. do. 65·32 do.

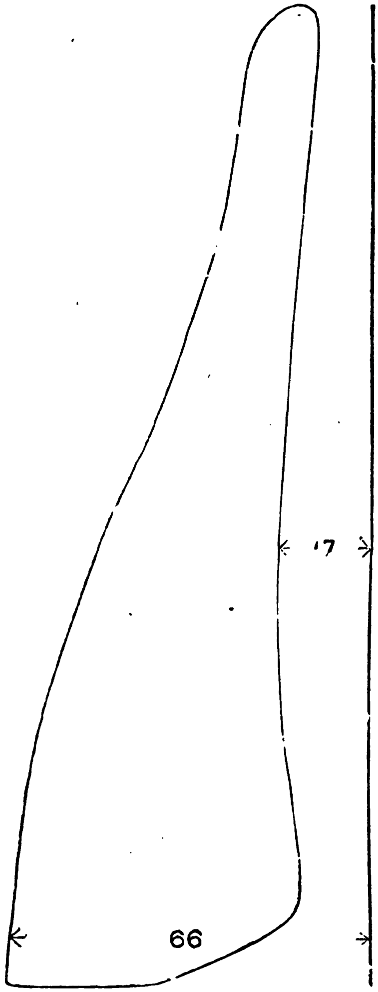
This is truly a startling conclusion; but if the data be correct, then it is inevitable.

One more fact is worthy of record. In some of the diagrams taken from these Engines, the initial pressure in the cylinders is found to be within half-a-pound of the boiler pressure for the time being.

Diagrams Nos. 47 and 48 are treated at such length in the chapter on compounding, that it would be altogether superfluous to say much here. They are from a Compound Engine with cylinders horizontal. The high pressure cylinder is 26 inches diameter, and the low pressure cylinder is 46 inches diameter, each 5 feet stroke of piston, and running 48 revolutions per minute, the cranks being placed at right angles one with the other, and therefore one piston is in the middle when the other is at the end of the stroke. The valves are of the long slide kind, and there are separate inlet

and outlet valves: the latter being underneath. The high pressure cylinder has a back slide cut-off valve. There is a receiver between the two cylinders, of about the capacity of the smaller one only, and

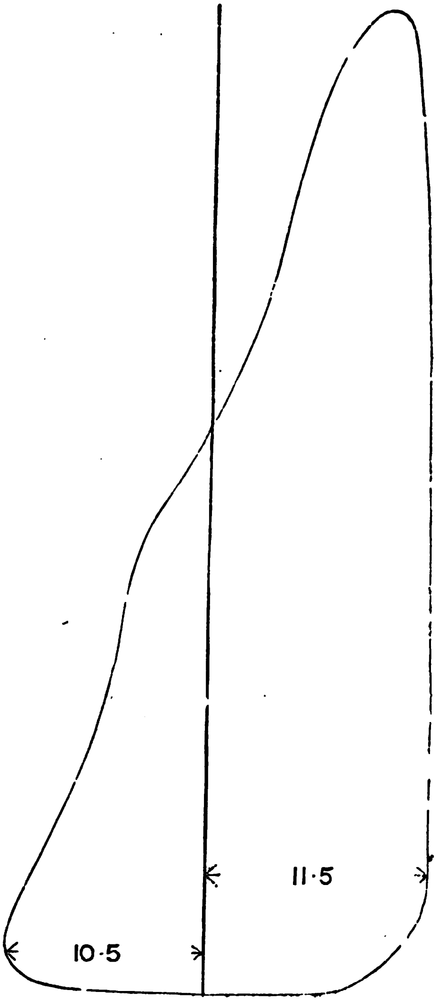
Diagram No. 47.



a pipe of very moderate dimensions; hence the greater back pressure in the middle of the stroke in the high pressure cylinder. The high pressure cylinder was formerly one of a pair, of equal dimensions, *and were then* worked as a pair of Simple Engines. Diagrams 13

and 14 are from one of these cylinders, when thus working. The boiler pressure was then, and still is, about 75lbs. For further information concerning these diagrams and Engines, the reader is referred to the chapter on compounding.

Diagram No.



CHAPTER X.

COMPOUNDING.

THE principle and practice of working Engines Compound may truly be said to have been contemporaneous with the growth and development of the Steam Engine; for a Compound Engine, invented by HORNBLOWER, was patented in 1781, just one year before any Steam Engine was made which gave a direct rotary motion by the arrangement of the crank and fly-wheel. Since that time Compound Engines have been made and used very extensively, though the number is only small in comparison with Simple Engines. During recent years the Compound Engine has been rapidly growing in favour because of the practical advantages derived from it,—enabling, as it does, owners of Engines of weak form and construction, by the application of a smaller cylinder for high pressure, to work the steam more expansively than would have been practicable with the one cylinder already in use. Great numbers of new Engines are also made on the compound plan—of various forms and proportions, as will hereafter be described.

It is still a moot point, amongst Engineers of the highest authority, whether the Simple Cylinder Engine, or the Compound Cylinder Engine is the more economical,—many eminent men being ranged on each side. The general experience of Engineers however, favours the adoption of the Compound Engine, because it is almost invariably found to give better results; but by what laws this fact may be accounted for, has not hitherto been satisfactorily determined. The question of compounding has been so far a mysterious and unsatisfactory study to the Engineer. The proportions of cylinders; their relative positions; the relative positions of the cranks and pistons; the positive and comparative amount of expansion in the high and low pressure cylinders;—these are questions which have hitherto been involved in the deepest obscurity.

In Compounding an Engine already in use the design will be governed greatly by the particular circumstances of the case. It would be impossible to give any general instructions which could be

universally applicable. The variety of arrangements is so great as almost to baffle description. There is first, the Beam Engine with two cylinders to one beam—HORNBLOWER'S,—generally called WOOLF'S plan, which has the cylinders close together at one end of the beam; the high pressure one being nearest to the beam centre, and as close to the low pressure cylinder as possible, so that the stroke of piston of the former is a little less than that of the latter,—ranging from three-fourths to four-fifths. The next, and best known plan of Beam Engine compounding, is the one originally patented by Messrs. ROBERTSONS, and afterwards re-patented by Mr. WM. MCNAUGHT, and now almost universally known by the term "McNaughted Engine." Indeed, so much is this term used in reference to compounding, that "compounding" and "McNaughting" are, with numbers, ignorantly supposed to be synonymous.

Sometimes two Beam Engines coupled together (that is a pair) are worked on the compound principle; the smaller one receiving its steam direct from the boiler, and thence passing into the large cylinder. Another very common plan of compounding is to have a pair of vertical Direct-acting Engines, with the fly wheel overhead, and a crank at each end of its shaft,—the cylinders being of the same length, but of different diameters. Still more common is the Horizontal Compound Engine, arranged on the same general principle as the last named.

There is also the Horizontal Compound Engine with the two cylinders in one line, but with some space between them,—the piston rods in some cases being jointed between the cylinders, so that the two pistons can be disconnected. Some Engines on this plan have the low pressure cylinder nearest the crank. Some of them are made with the air pump direct-acting horizontal—being indeed a continuation of it; and some with an L leg connected with the slide at the end of the piston rod at the back of the cylinder. Others are made with the air pump working parallel with the cylinder, but underneath, or at the side, a little below the level of the cylinder.

Then again, other Horizontal Compound Engines are made with the high and low pressure cylinders cast in one—being in one line,—the smaller cylinder being placed nearest the crank, and the larger cylinder having two piston rods passing outside the smaller cylinder, and being in the same plane as the high pressure piston

rod, and all the three being attached to one cross-head. The air pump is worked by an L leg connected by links to the cross-head, or by a connecting rod attached to the crank pin outside the connecting rod proper. This arrangement of cylinders has been practiced many years in Marine Engineering, and has during recent years been adopted in Engines for manufacturing concerns.

Another form of Direct-acting Compound Engine is that which now prevails in the large ocean steam ships. This arrangement consists in placing the larger cylinder vertically over the crank, and fixing the smaller cylinder on the larger,—the piston rod being in one piece, and passing through the low pressure cylinder into the high pressure one. There is invariably a pair of these in every steam ship where this class of Compound Engine is used,—many of them having the low pressure cylinders 96 inches diameter, and the high pressure cylinders 54 inches diameter each, with a stroke of piston of 5 feet, and running at a speed of 50 revolutions per minute,—being a speed of piston of 500 feet per minute, which is maintained day and night during the voyage of several thousand miles. Some vessels lately built have this class of Engines with low pressure cylinders 120 inches diameter, and high pressure cylinders 72 inches diameter, with a stroke of 5 feet 6 inches. These are amongst the largest Steam Engines ever made.

Next to be mentioned are the many varieties of ways of compounding Beam Engines other than the plans already described. The usual arrangement of the old make of Beam Engines is to have the fly wheel placed close by the wall, and its shaft continued through, with a spur wheel on the other side. It is now quite a common plan to place a horizontal, or slightly diagonal high pressure cylinder alongside this wall, and have a pin for the connecting rod fixed in the spur wheel at the proper radius. Or, if the spur wheel be close to the fly wheel, as is frequently the case, then a crank may be put on the end of the shaft instead of making the spur wheel serve the purpose of a crank as just described.

Another plan of compounding Beam Engines is to place the high pressure cylinder on a level with the crank, but at the opposite end of the engine house to the beam cylinders—the high pressure cylinder, as will be obvious, being horizontal, and its connecting rod attached to the same crank pin as is the connecting rod of the Beam Engine.

Many Beam Engines have been compounded by placing an oscilating cylinder in the position of the fixed horizontal cylinder just described, because sufficient room was not available for slides and connecting rod.

A Compound Engine which has been patented recently, and one of which has been erected in the neighbourhood of Huddersfield, deserves to be mentioned as an example of Engineering science in these days of scientific knowledge. The Engine referred to has three cylinders,—one high pressure, and two low pressure ones. The two low pressure cylinders are each single acting—are placed in one line,—so that one connecting rod serves for both pistons, and the outer end of each cylinder is open to the atmosphere. The patentee claims that by this arrangement he utilizes the pressure of the atmosphere, and thereby secures economy greater by such atmospheric pressure than can be obtained by one low pressure cylinder double acting. That men—and manufacturers too—can be led to believe such a statement, is the clearest possible proof of the necessity for the diffusion of a knowledge of the Steam Engine, and the laws involved in the working of it. Anything more absurd and childish than the claim advanced on behalf of this arrangement it would indeed be difficult to conceive.

In all the plans of compounding so far noticed, the pistons of the high pressure and low pressure cylinders have had a rigid connection, and beat at the same intervals of time, though not always simultaneously. There is still another class of Compound Engines extensively used, which consists of separate Engines, either singly or in pairs, and running either the same number of revolutions per minute, or having the High-pressure Engines running two or three to one of the Low-pressure Engines. In this arrangement the high pressure cylinders are almost invariably horizontal; and as invariably coupled by gearing with one or a pair of Beam Engines. It will be obvious that with this plan the high pressure cylinders will be some distance from the low pressure cylinders; and that the capacity of the pipes which convey the steam from the high to the low pressure cylinders, if of sufficiently large diameter, will usefully serve the purpose of a receiver. This, in addition to the fact of the High-pressure Engines making two or three revolutions for one of the Low-pressure Engines, will so equalize the back pressure, that diagrams from the High-pressure Engine will show a parallel line,

provided that the exhaust valves be properly set, and the ports sufficiently large.

It is important to notice here, that the initial pressure in the condensing cylinder should coincide, or nearly so, with the back pressure in the high pressure cylinder, unless too much condensation be taking place in the pipes, or some of the passages be too small to permit the free passage of the steam from one cylinder to another. All pipes, or steam vessels of any kind, through which the steam passes from the high to the low pressure cylinder, should be well covered with some good non-conducting material.

The arrangements above enumerated embrace all the best known types of Compound Engines, though there are many other variations in detail which it would be too tedious to particularise. In the arrangement of Compound Engines in which the cylinders have each a separate crank, but fixed on the same fly wheel shaft, and therefore having the pistons beating at the same intervals of time, a very knotty and much contested point has been, as to what angle the cranks ought to be placed relatively to each other. This will be best determined by the particular circumstances of each case. Where the two cylinders are cast in one, or are placed as near together as possible, side by side and parallel with each other, and where the steam passes direct from the high to the low pressure cylinder, and has only the distance from cylinder to cylinder to traverse,—then the cranks should be placed at nearly opposite points, the low pressure crank leading by one-twelfth or one-tenth of the stroke, because the exhaust valve of the high pressure cylinder should open when within that distance of the end of its stroke, and the low pressure cylinder should necessarily be ready to receive the steam. In Engines having the cylinders in parallel lines, but several feet apart, and where the exhaust steam from the high pressure is conveyed to the other cylinder by one pipe—as is invariably done,—the circumstances are so much changed, that the cranks may be placed at other angles. With a receiver between the cylinders, it would undoubtedly be best to have the cranks placed at right angles one with the other, as it would give greater regularity of motion to the machinery,—which is always a desideratum.

The theory of the Compound Engine in relation to the law of expansion, and the comparative efficiency and economy of the Simple and the Compound Engine has not yet been discussed. We will now

examine this part of the subject in its various phases, and show what are the principles involved, and what are the conclusions to be drawn from them. We will also show how to determine the proportions which should exist between the high and low pressure cylinders, in order to obtain the greatest effective energy of the steam. These are questions of the highest importance in Steam Engineering.

Theoretically, there is no difference in the expansive power of a given quantity of steam, of given initial and terminal pressures, whether it be effected in one or two cylinders,—provided that the Compound Cylinders be correctly proportioned, &c., as will hereafter be shown; otherwise a positive loss will result from compounding. The theoretical advantage is greatly in favour of the Single Cylinder Engine, in comparison with nearly all the Compound Engines now in use; practically, however, there is a decided advantage in the compound principle when applied to most of the Engines now working. The advantage derived from the principle of the Compound Engine is chiefly this: It is a mechanical arrangement whereby the extremely variable pressure of the steam is more equably distributed through the revolution of the crank, when a high degree of expansion is obtained.

If a very high degree of expansion should be attempted in one cylinder—as Engines are usually made,—the results would be disastrous. Suppose, for instance, that we have steam of 120lbs initial pressure (105lbs × 15lbs vacuum = 120lbs) expanded to 7·5lbs terminal pressure, which is 7·5lbs below the atmospheric line (assuming the atmosphere to be equal to 15lbs per inch, = 30·56 inches of mercury). Thus we shall have 120 ÷ 7·5 = 16; that is, the cut-off will be at one-sixteenth of the stroke. Practically it would be much earlier than one-sixteenth of the traverse of the piston to give the requisite measure of steam, because there is the clearance of piston and the capacity of ports and thoroughfares—the whole fulcrum capacity—to fill; and if this space be added to one-sixteenth of the actual traverse of the piston, the terminal pressure would then be higher than 7·5lbs, as given above, by the proportion which such fulcrum capacity bears to that given by the traverse of piston to the point of cut-off—basing the calculation on the law of MARRIOTTE. We will suppose the cylinder to be 800 inches area, and the piston speed to be 500 feet per minute. Then 120lbs per inch initial pressure, *minus* 2lbs back pressure (= 18lbs vacuum)

will give 118lbs per inch effective initial pressure on the piston. Then the area of piston being 800 inches × 118lbs pressure = 94,400lbs, which is the pressure on the piston when the crank is on the dead centre. This enormous pressure (nearly 42 tons) is brought suddenly on the piston; and unless all the reciprocating and other parts of the Engine be of a commensurate weight and strength, the violent and useless pressure on the crank pin and crank shaft journal, when on the dead centre, will be mischievous, as may be easily seen.

Let us now see what would be the corresponding effect of expanding steam to the same degree in two cylinders compounded. For 120lbs initial pressure, and 16 expansions, we will assume the high pressure cylinder to be one-eighth of the area of the low pressure cylinder, and of the same length, so that the high pressure cylinder will be 100 inches area, and the low pressure cylinder 800 inches area, as in the Single Cylinder Engine; and we will use the same measurement and pressure of steam as before. As the cut-off in the single cylinder was one-sixteenth of the stroke; and as the high pressure cylinder of the Compound Engine is one-eighth the area of the low pressure; then, by cutting off at half stroke in the high pressure cylinder, exactly the same measurement of steam has been used. We shall now have 120lbs initial pressure, cut off at half stroke = 60lbs terminal pressure in the high pressure cylinder. If we now cut off the steam in the low pressure cylinder at half stroke, then it is evident that the steam of 60lbs terminal pressure in the high pressure cylinder must now pass into a space of four times the capacity of the high pressure cylinder, and therefore the pressure will be reduced from 60lbs to 15lbs—or from 45lbs above the atmosphere to the atmospheric line,—thus giving a back pressure on the high pressure piston of 15lbs, or simply atmospheric pressure. Then with 15lbs in the low pressure cylinder, cut off at half stroke, the terminal pressure will be 7·5lbs as in the case of the Single Cylinder Engine just considered.

Now let us see what will be the pressure on the crank pin at the commencement of the stroke, assuming the two cylinders to be in one line, and therefore the total effective initial pressure on both pistons communicated through one connecting rod, and to one crank pin. The high pressure cylinder, with an area of 100 inches, and *120lbs initial* pressure, minus 15lbs back pressure = 105lbs effective

initial pressure. Then, 105lbs × 100 inches area = 10,500lbs on the high pressure piston. Low pressure cylinder = 15lbs initial pressure — 2lbs back pressure = 13lbs effective pressure on the piston. The area of the low pressure cylinder being 800 inches × 13lbs = 10,400lbs. Here, then, we find that the total initial effective pressure on both pistons amounts to 20,900lbs; whereas, in the single cylinder with the same pressure of steam, and an equal degree, of expansion, the pressure on the piston amounted to 94,400lbs, or more than 4½ times the amount of the pressure on both the pistons of the Compound Engine. If the pressure on the pistons of the Compound Engine had been distributed through two cranks, as is commonly done, then the difference between the single cylinder and the Compound Engine would be still greater in respect to the amount of pressure on the crank pin.

Let us now see what would be the pressure on the crank by cutting off the steam in the high pressure cylinder as before, at half stroke, and in the low pressure cylinder at quarter stroke. Then the high pressure cylinder with 120lbs initial pressure, cut off at half stroke, will give 60lbs terminal pressure. The low pressure cylinder being eight times the area, and cut off at one quarter stroke, will equal twice the capacity of the high pressure cylinder, which the steam must fill in passing from one to the other, and therefore the terminal pressure of 60lbs in the high, will be reduced to 30lbs in the low; which being cut off at one-quarter stroke, will give 7·5lbs terminal pressure. We shall now arrive at the following result:

High pressure Cylinder = 120 — 30 = 90lbs
Low ditto ditto = 30 — 2 = 28lbs

High pressure 90lbs × 100 inches area = 9,000lbs
Low ditto 28lbs × 800 ditto = 22,400lbs

Total pressure on both pistons = 31,400lbs

It will be seen that by merely changing the point of cut-off in the low pressure cylinder from one-half to one-quarter stroke, the effective initial pressure on the two pistons has been increased in the ratio of two to three.

Sufficient has here been said to enable the reader to understand this part of the subject.

We will now calculate the comparative power of Simple and Compound *Engines, the* latter variously proportioned, and as com-

monly found working. Let us first take the Simple Engine (single cylinder), and adopt the dimensions already given,—say 800 inches area, and 500 feet per minute speed of piston. We will also take the same initial pressure as before, $105 + 15 = 120$lbs absolute pressure. Now, 120lbs initial pressure, cut off at one-sixteenth, = 7·5lbs terminal, and 28·29lbs average pressure; and deducting 2lbs back pressure (= 13lbs vacuum), the average effective pressure on the piston will be 26·29lbs. Then—

$$\frac{800 \text{ inches} \times 26{\cdot}29\text{lbs} \times 500 \text{ feet}}{33{,}000} = 318{\cdot}664 \text{ H.P.}$$

Let us now calculate what will be the power given by a Compound Engine using the same quantity of steam at the same pressure. Let the low pressure cylinder be 800 inches area, as before, and the high pressure cylinder 100 inches area, being one-eighth of the area of the low pressure cylinders, which is a very common proportion now in use for such pressure of steam. Let the steam be cut off at half stroke in each cylinder. The high pressure cylinder having 120lbs initial pressure, and being cut off at half stroke, will give 60lbs terminal pressure, and 101·58lbs average pressure. As the steam must now be transferred to the low pressure cylinder, where it is cut off at half stroke, its volume will be increased four-fold, and its pressure reduced in the same ratio, so that we shall now have 60lbs $\div$ 4 = 15lbs initial pressure in the large cylinder, which, being cut off at half stroke, there will be a terminal pressure of 7·5lbs, and an average pressure of 12·6975lbs. As the initial pressure in the low is 15lbs, so the back pressure in the high pressure cylinder will be 15lbs also. The result may now be stated thus: High pressure cylinder 101·58lbs — 15lbs back pressure = 86·58lbs average effective pressure; low pressure cylinder, 12·6975lbs — 2lbs back pressure = 10·6975lbs average effective pressure on the piston.

HIGH PRESSURE CYLINDER.

$$\frac{\text{Area, } 100 \text{ inches} \times 86{\cdot}58\text{lbs} \times 500 \text{ feet}}{33{,}000} = 131{\cdot}18 \text{ H.P.}$$

LOW PRESSURE CYLINDER.

$$\frac{\text{Area, } 800 \text{ inches} \times 10{\cdot}6975\text{lbs} \times 500 \text{ feet}}{33{,}000} = 129{\cdot}66 \text{ H.P.}$$

Total of both cylinders = 260·84 H.P.

As 260·84 : 318·664 :: 81·85 : 100. Thus we find that by the above proportions of cylinders, the total power of the Compound Engine is only 81·85 per cent. of that of the Simple Engine, with exactly the same quantity and pressure of steam.

Let us now ascertain what would be the amount of power with the same initial pressure of steam, and the same dimensions of cylinders, but having the steam maintained at a uniform pressure throughout in each cylinder. Now we shall have 120lbs pressure expanded into a cylinder of eight times the capacity = 15lbs pressure in the low pressure cylinder, and this will also be the back pressure in the high pressure cylinder; this being deducted from the absolute pressure of 120lbs = 105lbs effective average pressure. Allowing a back pressure of 2lbs in the low pressure cylinder as before, we shall have 15 − 2 = 13lbs average effective pressure on the piston. The result will be as follows—

HIGH PRESSURE CYLINDER.

$$\frac{\text{Area, } 100 \text{ inches} \times 105\text{lbs} \times 500 \text{ feet}}{33{,}000} = 159 \cdot 09 \text{ H.P.}$$

LOW PRESSURE CYLINDER.

$$\frac{\text{Area, } 800 \text{ inches} \times 13\text{lbs} \times 500 \text{ feet}}{33{,}000} = 157 \cdot 57 \text{ H.P.}$$

Total of both cylinders = 316·66 H.P.

In this case double the quantity of steam has been used that was used in any of the cases previously given, and yet a less calculated power has been obtained than was found to be given by the Simple Engine expanding 16 times.

We have already seen what is the power developed by a Compound Engine with 120lbs initial pressure, and 7·5lbs terminal pressure = 16 expansions, the ratios of cylinders being as 1 : 8, and the steam cut off at half stroke in each case. Let us now see what will be the result of using steam at 60lbs initial pressure, and 7·5lbs terminal pressure = 8 expansions, with the same cylinders, —the steam in the high pressure cylinder being kept on at full pressure during the whole stroke, and cut off in the low pressure cylinder at half stroke. As the terminal pressure in the high pressure is 60lbs, and as it must thence pass into the low pressure cylinder of eight times the capacity, but in which it is cut off at half stroke, it will equal four times the capacity of the high pressure

cylinder, and will therefore have a pressure of $60 \div 4 = 15$lbs to the point of cut-off, and a terminal pressure of 7·5lbs. The initial pressure in the low pressure being 15lbs, the back pressure in the high pressure cylinder will also be 15lbs.

The high pressure cylinder will have an average effective pressure of 45lbs ($60 - 15 = 45$); and the low pressure cylinder with an initial pressure of 15lbs, and cut off at half stroke, will have an average of 12·6975lbs; and deducting 2lbs back pressure, the average effective pressure on the piston will be 10·6975lbs. The result for comparison with the other cases can now be presented.

HIGH PRESSURE CYLINDER.

$$\frac{\text{Area, 110 inches} \times 45\text{lbs} \times 500 \text{ feet}}{33{,}000} = 68{\cdot}18 \text{ H.P.}$$

LOW PRESSURE CYLINDER.

$$\frac{\text{Area, 800 inches} \times 10{\cdot}6975\text{lbs} \times 500 \text{ feet}}{33{,}000} = 126{\cdot}63 \text{ H·P.}$$

$$\text{Total of both cylinders} = 194{\cdot}81 \text{ H·P.}$$

This gives 61 per cent. of the power developed by the Simple Engine (single cylinder) with 120lbs initial pressure, and 16 expansions, and yet the same quantity of steam has been used in each case.

The cases above given show clearly the extremely variable results which may be obtained from a given quantity of steam, and also by a given amount of expansion. The variation in practice is even greater than these suppositious cases. The calculations show the supreme importance of the proportions and arrangements of cylinders of Compound Engines. Most Engineers have so far been groping in the dark, with no true principle to guide them; hence we have such immense diversity in practice—ranging, in the proportion of cylinders from 1 to 1 (equal capacity) to 1 to 10, with every intermediate proportion between these two extremes —all which proportions we have seen actually at work.* The knowledge of a true principle in Compounding Engines has long been much wanted to enable all who are adopting them to know exactly what is best to be done to obtain the highest duty from a given amount of steam.

*A case has come to our knowledge in which the cylinders of a Compound Engine are in the ratio of 1 : 11¼.

We have seen that Compound Engines do not at present give the theoretical duty of Simple Engines with the same initial and terminal pressures. The law of proportion has now been determined by our Mr. Wm. SUTCLIFFE, by which the Compound Engine will give the same theoretical duty as the Simple Engine—that is, according to the law of MARRIOTTE, on which our theoretical calculations are usually based.

We will now give the rule founded on the law of proportion just named, by which Engineers in all future time will be able at once to determine the exact proportions of cylinders, &c., required in every case.

RULE FOR FINDING THE PROPORTION OF CYLINDERS AND THE POINT OF CUT-OFF FOR COMPOUND ENGINES OF TWO CYLINDERS.

Ascertain the initial and terminal pressures at which it is intended to work the steam. Then divide the initial by the terminal pressure, and the quotient will be the number of times which the steam has been expanded. Then get the square root of the number of expansions;* and as the root is to the square, so must be the area of the high pressure cylinder to that of the low pressure cylinder.

Then, again; as the root is to the square, so must be the point of cut-off in each cylinder.

This will be found in all cases to give the required degree of expansion; and also to give the highest aggregate effective pressure —therefore the best attainable results. Furthermore, it will give an equal absolute initial pressure on each piston—the difference in the effective initial, and effective average pressures, being found by multiplying the number of pounds back pressure in each cylinder by its area in square inches.

We will now give an illustration of cylinders and diagrams based on this law of proportion, with the calculations which prove the accuracy of it,—making it clear to Engineers who understand theoretical diagrams.

A short explanation will be desirable in order to make this illustration clearly intelligible to everyone.

* The square being the number of expansions. The expression "Number of Expa sions," whenever used in this work invariably means the number of volumes to which a given initial volume of any given pressure is, or may be, expanded by the law of inverse ratio of volume to pressure.

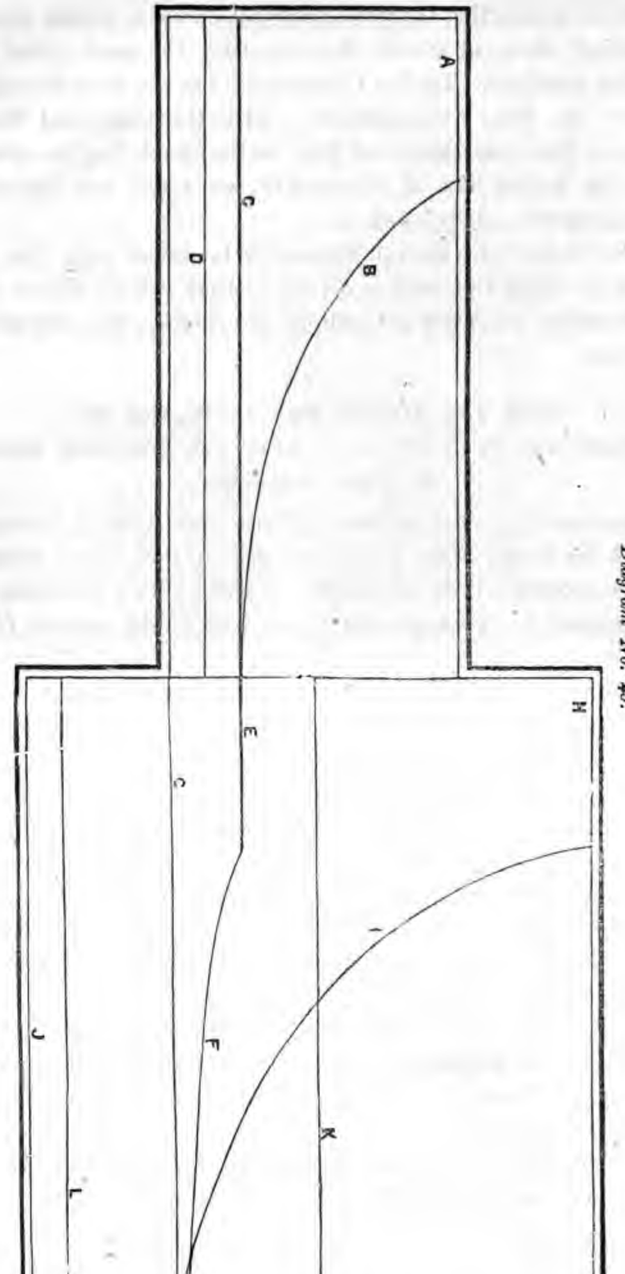


Diagram No. 49.

Of the enclosing lines, the inner and fainter of them represent the relative diameters of the two cylinders of a Compound Engine, the areas of which are in the ratios of 1 : 4, being the proportions given by rule for 16 expansions, as will be found in column H in the table on compounding, which will be given hereafter. The various parts of this illustration will most easily be understood by keeping in view column H just referred to.

Letter A in the illustration shows the steam line, or initial pressure in the high pressure cylinder, which is here 120lbs,—being to a scale of 80lbs per inch; B is the expansion line; and as the steam is cut off at one quarter of the stroke, the terminal pressure, is 30lbs. C represents the back pressure, which is also 30lbs. D represents the atmospheric line in the high pressure cylinder. E represents the initial pressure in the low pressure cylinder as a continuation of the terminal pressure in the high pressure cylinder, and on the same scale of 80lbs per inch. F represents the expansion line, and G the line of absolute vacuum on the same scale. H represents the steam line or initial pressure in the low pressure cylinder,—being to a scale of 10lbs per inch, as are all the lines indicated by the remaining letters. I represents the expansion line in the low pressure cylinder; and as the initial pressure here is 30lbs, and the cut-off one-fourth, the terminal pressure is 7·5 lbs. J represents the line of absolute vacuum in the low pressure cylinder. K represents the atmospheric line, and L the back pressure line, which is 13lbs below the atmospheric line.

As the cylinders are in the ratio of 1:4; and as the cut-off in the latter is at one-fourth the stroke, it is evident that to this point it is of the same capacity as the whole capacity of the former; therefore the larger cylinder will have the same pressure to the point of cut-off as the terminal pressure in the smaller,—omitting here any loss of pressure which may occur by the passage of the steam from one cylinder to another, by reason of friction, smallness of ports and thoroughfares, &c., which will be considered a little further on.

To produce diagrams approaching these in form, will require, as every Engineer knows, efficient cut-off valves; and it will be further necessary to have a *receiver* between the cylinders; otherwise the back pressure in the high pressure cylinder will be very variable. With a receiver, of say 10 times the capacity of the high pressure cylinder, the back pressure there could only vary by a very small

amount. In the case of one corresponding to illustration, where the pressure between the cylinders is 30lbs, the variation will only be $2\frac{1}{4}$lbs due to compression in the receiver. If we examine a case of 8 expansions, and suppose the receiver to have 10 times the capacity of the high pressure cylinder, then, with an initial pressure of 60lbs, the receiver would have a pressure of 21·21lbs, and the variations in pressure due to the cause stated above would be less than $1\frac{1}{2}$lbs.

Every Engineer who has had much experience with Compound Engines knows that there is usually a difference of a few pounds in pressure, between the back pressure in the first cylinder, and the initial pressure in the second. A large proportion of such loss of pressure is due to causes which have been described elsewhere. The loss is very small in the best constructed Compound Engines. With efficient valves, thoroughfares, and passages; with the cylinders steam jacketed, and with the steam in the receiver superheated,—it is highly probable that very little, or no fall in pressure would occur.

By the rule of proportion given, the same co-efficient of expansion is obtained as when the same number of expansions is effected in a single cylinder,—and this is the highest attainable. It is clear then, that the superiority (if any) of the Compound over the Simple Engine must be determined by some other cause than merely the expansion of the steam. That cause yet remains in the realm of debatable questions. The loss of power as shown above, in various proportions and arrangements of cylinders which give a less co-efficient of expansion than the single cylinder, or properly proportioned Compound Engine, is not the only loss which is often found. A very frequent source of loss is found in the passage of the steam from one cylinder to another.

In addition to the advantages shown to result from the expansion of steam in both cylinders, there is another important advantage which is not so obvious, but which nevertheless is as real as that of expansion, though it does not admit of such direct and simple demonstration. If the reciprocating parts of an Engine had no weight, then we should require the steam to be kept on at an equal pressure during the whole length of the stroke, in order to obtain uniformity of pressure on the crank pin. Fortunately, this is not so, or we *should not be able* to obtain, to the same extent, the benefit of the *law of expansion.*

When the piston is at the end of the stroke it is in a state of rest, and the inertia of this state of rest has to be provided for. The amount of pressure required for this purpose will be determined by the weight of all the reciprocating parts, and the acceleration of speed which is to be given to them. The laws of inertia equally require that the full propelling power should not be continued past the point of highest velocity in any case where the speed is considerable, or even moderate. The greater the speed of piston, and the weight of the reciprocating parts, and the less will be the danger of high pressure and early cut-off, because the pressure of steam required to overcome the inertia of rest, will relieve the pressure on the crank pin at the commencement of the stroke, and add to the pressure there during the latter part of the stroke, when the pressure of steam is low; and if the reciprocating parts be of commensurate weight, the impulsive and momentive forces may be approximately balanced throughout the stroke, but only within a very limited range of expansion. From the laws of motion it follows that whatever force may be communicated to the piston and other reciprocating parts to overcome the inertia of rest, and to give an acceleration of speed, will as certainly be given out again during the latter part of the stroke. These remarks apply exclusively to the Steam Engine with a revolving motion of crank, and having a connecting rod attached, giving a differential motion to the piston.

Even before the Steam Engine was made to give a rotary motion by means of the crank, and when it was used merely for pumping water, this law of inertia was practically made manifest: it was found that by keeping the steam on the piston during the full length of the stroke, the weight of the moving parts acquired a momentum which accelerated their speed, and had to be violently arrested at the end of the stroke. Experience soon showed that the steam, at the low pressure then used, might be cut off at one-third of the stroke with great advantage,—first, in securing a more uniform motion of the piston and pump, and second, in economy of steam. The piston of an Engine, the connecting rod of which is attached to a crank to give a rotary motion, is subject to the same laws; and the amount of pressure required to start it, and the power given out in arresting it, will be governed by the weight and speed.

Now these facts and reasons lead inevitably to the conclusion that there will be a *decided* advantage in cutting off the steam in

each cylinder in the way shown by the law of proportion which we have given. If the full pressure of steam should be continued too far, then the reciprocating parts would acquire a momentum in excess of the requirements, which, being increased by the higher terminal pressure of steam relatively to the initial pressure, would have an injurious effect in impelling the mass in motion with too great a force against the crank pin when at and near the termination of the stroke. This fact involves a corelative one, viz: that the pressure at the commencement of the stroke is relatively too low. If, on the other hand, the pressure at the commencement of the stroke be great, and the cut-off very early, the impulsive force may not continue long enough, as is indeed the case in a great number of Engines. So common is this fact, that anyone acquainted with the working of the Steam Engine can discover the irregularity of motion arising from this source, by listening to the machinery, whereby every beat of the piston can be detected. By an increase of the speed of the Engine, with the same pressure and point of cut-off, this would be rectified, because the fly wheel, by its increased speed, would have the sum of its momentum increased. If the inertia of the fly wheel be great enough, it will equalise almost any possible difference of pressure on the crank pin, so that no perceptible variation of speed will occur.

There is another part of the question of the Compound Engine embraced in the Rule of Proportion, which is too important to be overlooked. The foregoing remarks on inertia and momentum apply to the reciprocating parts of the Steam Engine, without special reference to the Compound Engine. But it will be self-evident as an axiom in geometry, that whatever amount of expansion is best for one cylinder must be best for the other also; hence it will be manifestly the best to cut off the steam in both cylinders at the same point. By this arrangement greater uniformity, smoothness, and regularity of motion will be obtained, because the pressure on the crank pin will be more uniform throughout every part of the circle. Now, wherever force, or pressure, is improperly applied, some portion of it at least must be wasted. For, although the sum of the forces is the same, however distributed, yet by the improper application, some part of the force will be lost, so far at least as not producing a beneficial result.

Take for example an Engine of the ordinary speed of 30 to 40 *revolutions* per minute. To have a high initial pressure and very

early cut-off, would be to apply a pressure very far in excess of what would be required to overcome the inertia of the reciprocating parts, and the average pressure on the crank pin requisite to maintain uniformity of motion of the fly wheel; and therefore, the excess of pressure above these requirements, would, when the crank is at, and near, the dead centre, be absorbed by the crank shaft and bearing, and could not produce the most useful effect. In this case there would be so large a proportion of the pressure exerted on the journal of the crank shaft relatively to that exerted in the direction of rotation, that some of the energy will inevitably be dissipated. If, on the other hand, the pressure of steam on the piston be too great during the latter part of its stroke, the momentum at its termination will be too great, and therefore some portion of the force will be wasted, or dissipated. Another evil arising from this excessive terminal momentum of the reciprocating parts, will be an uneven, jarring, and unbalanced motion of the Engine and gearing, which is highly objectionable in every way, and will have an injurious effect on every part of the Engine, as tending to dangerous strains and ultimate breakdowns.

The reasons already advanced are sufficient to prove the importance of a given point of cut-off in the cylinders of Compound Engines; and by parity of reasoning, of cutting off at the same point in each cylinder when the pistons travel at the same speed, so as to have the impulsive force on the crank pin distributed equally through every degree of its revolution, or as near this as is possible by the mechanical arrangement of the Steam Engine.

A very important question in respect to the Compound Engine has yet to be determined. We have seen that the theoretical results, calculated by the law of MARRIOTTE, of the Simple and the Compound Engine, are equal when the latter is proportioned and arranged according to the law of proportion for Compound Engines which we have discovered. Yet, with proportions and arrangements which theoretically give far worse results, we find that practically greater economy is obtained;—or apparently so, for the question is, perhaps, not finally settled.

Now, assuming for the present that the Compound Engine is *really* more economical than the Simple Engine using the same measure and pressure of steam, it is very desirable that we should know the cause, or causes, so that our practice may be based on

clearly ascertained principles, instead of that empiricism which has often mistaken coincidence for cause and effect. Whatever may be the causes which tend to the greater economy of the Compound over the Simple Engine, it is highly important that they should be ascertained and their relative values determined.

A theory has been propounded by an American Engineer, Mr. C. E. EMERY, of New York, that a greater amount of condensation takes place when a given quantity of steam is used in one cylinder than when used in two cylinders compounded. This greater condensation is supposed to result from an assumed law of the amount of condensation being as the square of the difference of temperature. This theory is believed by numbers, and yet there is a singular absence of evidence in support of it. As our great experimentalists and writers on this subject seem to be unaware of such a law, we must defer coming to a final conclusion until the question has been determined by exact, careful, and well authenticated experiments. Professor W. J. M. RANKINE does not favour such a theory, but states that the amount of condensation is as the difference of temperature within the practical range of a Steam Engine.

Another theory advanced, and advocated by some Engineers, is known as the "Heat Trap" theory, which assumes that as only one cylinder is in communication with the condenser a less total amount of condensation occurs. On this theory professor RANKINE expressed his opinion in a communication to the *Engineer*, dated June 10th, 1872 (only six months before his death), in the following words :

"It is obvious that no rule of universal application can be laid down as to whether the preference is to be given to fewer cylinders and greater journal friction, or to more cylinders and less journal friction; but that each case must be decided according to its own special circumstances to the best of the judgment of the Engineer.

"As for what you call the 'Heat Trap' action, it undoubtedly takes place with an economical effect, where neither jacketing nor sufficient superheating is used; but the use either of the jacket or of sufficient superheating causes that action to disappear, or to become quite unimportant."

Careful experiments and investigations which we have made, seem to prove that less heat passes into the condenser, and therefore less into the cylinder, for a given amount of power with the *Compound Engine.* The increase of temperature given to the injection

water may be considered a reliable test of the amount of heat passing through the cylinder—all other things being equal. Amongst the numbers of cases investigated the results are found to be very variable.

The most noteworthy instance of Compounding a Simple Engine which we have met with is in the vicinity of Manchester. About five years ago (in 1869) a pair of new Horizontal Engines were erected and started. The cylinders were each 5 feet stroke, 26 inches diameter, and 48 revolutions,—the cranks being placed at right angles one with the other. The initial pressure was usually 65lbs above the atmospheric line, and sometimes 70lbs. Diagrams Nos. 13 and 14 are from one tap of one of these cylinders, and are fair representatives of the set. The pressure at the point of release (opening of the exhaust) is seen to be on the atmospheric line. The co-efficient of expansion is found to be 2·2.

A little more than a year ago the Engines were compounded, by simply removing one of the cylinders, and putting a new one in its place, of the same length, and 3·1 times the area, so that now the cylinders are,—high pressure 26 inches diameter, and low pressure 46 inches diameter, 5 feet stroke, and 48 revolutions per minute. A receiver of about the capacity of the smaller cylinder is placed between the two. The initial pressure in the high pressure cylinder is now 65lbs as before (80lbs absolute), and the terminal pressure at the point of release in the low pressure cylinder is 10lbs absolute pressure. Diagrams 47 and 48 are from the high and low pressure cylinders respectively. The point of release is at the same distance from the end of the stroke as in the case of the Diagrams Nos. 13 and 14 from the Simple Engine. The increase of temperature given to the injection water before compounding was (so the Engineer informs us) usually about 50°, sometimes a little more, and sometimes a little less; whilst since compounding, the increase of temperature has been only about 35°. Both the condensers which were used for the two cylinders of the Simple Engines are now used for the condensing cylinder of the Compound Engine.

The cylinder capacity being now 1·55 times greater than before, and the initial pressure the same, there is a greater ratio, and a higher co-efficient of expansion. The actual gain here, as shown by diagrams, *amounts to 13* per cent. As the terminal pressure is now

lower, so is the temperature, which at 10lbs pressure = 193°; and as the amount of vacuum since compounding has been 11·5lbs, whilst it was 12lbs before, the temperature in the cylinder is not reduced so low. Taking the pressures as indications of temperatures, we shall arrive at the following result:

BEFORE COMPOUNDING.

Terminal pressure (at point of release)	15lbs =	213°
Vacuum line	3lbs =	141·6°
Difference..........................		71·4°

SINCE COMPOUNDING.

Terminal pressure (at point of release) =	10lbs =	193°
Vacuum line................................ =	$3\frac{1}{2}$lbs =	148°
Difference..........................		45°

Assuming what is shown by calculation, that for the same power 18 per cent. less water in the state of steam passes through the cylinders of the Compound Engine, then it is evident that a much larger sum of heat is given up by the steam to the condenser in the case of the Simple Engines than in the Compound Engine,—independently of any occult cause of condensation which may be supposed to occur. It should be noted here, that the Diagrams Nos. 47 and 48, from the Compound Engine, show a greater amount of power than Diagram No. 13, from one tap only of the pair of cylinders of the Simple Engine, as this one is below the average of the set; so that the quantity of steam and its temperature, which passes into the condenser in the case of the Simple Engine, will be slightly in excess of what is given above.

It must not be supposed that the difference of temperature of the steam at the point of release, and at the vacuum, or back pressure line, is sufficient to account for the whole of the increase of temperature given to the injection water. A large proportion of such increase is the result of the liquefaction of the steam, whereby its latent heat is rendered sensible. What proportion of the latent heat is thus rendered sensible can only be determined by very exact and refined experiments. It is quite evident that only a portion of the steam is liquefied, otherwise the sum of heat given to the injection water would be nearly the same for a given weight of steam, whatever might be the temperature at the point of release.

That a greater amount of heat is transmitted to the condenser by the Simple Engines, and for the same weight of steam, as found by the volume and pressure at the termination of the stroke, is certain, as will be clearly shown further on. In the absence of experimental tests we have no alternative but to assume the temperature as represented by the pressure. On this assumption the sum of heat which passes from the Simple Engines to the condensers will be greater than from the Compound Engine, because a larger quantity of steam is consumed; the difference in the two cases is in the proportion of steam condensed in the cylinders, and which is in the state of water in suspension in the steam at the termination of the stroke. Such water will necessarily have a higher temperature as the pressure is higher; but whether the difference is greater than that which corresponds to the pressures, must for the present be left an open question.

The saving in fuel by the change from the simple to the compound principle has been in this case perfectly marvellous. The consumption of coal per week before the change was 59 tons, whilst since it has not been more than 42 tons,—the boiler pressure, and every other condition remaining as before. The weight is a little more than 500 I.H.P. This is a very striking example of the beneficial results of compounding. It is important to state that neither of the cylinders are steam jacketed, nor were they before the change. It is rarely that we have the privilege of examining a case of changing from the simple to the compound principle, where all other conditions remain unchanged as in this case, and especially where the initial pressure and ratio of expansion are so great. Its value, as a criterion by which to judge of the merits of the compound principle, is great accordingly.

Though the proportion of fuel here saved may seem great, yet the real proportion is still greater. It must be remembered, that of the total consumption above given, an uncertain, though large amount, is required for warming the mill—which is a shed for cotton spinning only,—and for maintaining the fires and other requirements, when the Engine is not running; and this amount remains a constant quantity whatever may be the efficiency of the Engine. This consumption we will estimate at eight tons per week, which cannot be considered excessive. When we deduct this amount *from the former consumption* of 59 tons, then the amount used for

power alone would be 51 tons per week. As the reduction in the consumption by compounding has been 17 tons per week, then it is clear that the saving for power only is one-third, or 33 per cent. Of this proportion, 13 per cent. is due to the greater co-efficient of expansion, which in the Compound Engine is 2·55, being 13 per cent. greater than in the single cylinder expansion. When these are corrected by the method for obtaining the true co-efficient, which will be explained in the latter part of this chapter, the gain here will be nine per cent. There remains now a balance of 24 per cent. clearly due to the principle of compounding.

Assuming the evaporative efficiency of the boilers at 7lbs of water per pound of coal, the terminal pressure of steam in the Compound Engine, calculated for the weight of water contained in the state of steam, would show a consumption of coal of 29 tons for 60 hours, which is the time run per week. As by the calculations already given, the consumption for power is 34 tons, then it would appear that nearly 15 per cent. of the steam is condensed in the cylinders, and passes out on its way to the condensers in the state of water. Before compounding, the condensation, computed by the same method, must have amounted to 43 per cent. Both include condensation due to work done by the steam, and by contact with the metal of the cylinder, in excess of that which may have been re-evaporated when the piston is at the point of release of the steam.

It will be seen from the details given in connection with this case how difficult it is to determine exactly and absolutely the causes which operate in securing such valuable results. It is quite manifest that something remains yet to be done to raise the science of steam engineering out of the realms of darkness. Any number of examples of changing from a Simple to a Compound Engine by which a saving of fuel has been effected, could be given; but as the particular facts and circumstances connected with them are either wholly wanting, or do not serve as a sufficiently exact comparison, they are of little value for our present purpose, which is, to give such data as may lead to a solution of the problem.

To make an exact comparison between the Simple and the Compound Engine, many precautions should be observed. All the conditions should be the same. There should be the same boiler, or *boilers, the* same kind of coal, the same method of firing, the same

boiler pressure, the same initial pressure in the cylinder, the same volume admitted, the same amount of vacuum or back pressure in the condensing cylinder, the same machinery and weight on (lifting water would be the most accurate test),—the same quantity and temperature of injection water, &c. The experiment would best be made on a Compound Engine having the high and low pressure cylinders in one line, with a space between, and in which the piston rods are joined by a socket and cotter. The Engine, or Engines, might thus be run simple and compound alternate weeks or months, and the results noted. All the cylinders should be steam jacketed, so that both principles could be tried *with* and *without* steam in the jackets.

As a further means of ascertaining the causes of the difference between the two principles of working, it is desirable to make provision for collecting all the water passing out of the cylinders which may have condensed there, and ascertain its weight; and also to have a thermometer inserted into the exhaust pipe, as near to the cylinder as practicable, in order to ascertain the temperature of the exhaust steam. By adopting surface condensers, the whole of the steam consumed could easily be ascertained by collecting all the water condensed, and measuring or weighing it. This would give us a complete and perfect test of the relative economy of the two principles. Furthermore, it would assist in coming to a clearer knowledge of the causes operating, if a few thermometers were suspended in different parts of the Engine house, and the temperature shown by each noted at specified times every day,—the temperature of the external atmosphere in the shade being noted also at the same time. It would be well also to ascertain the temperature of the metal near the crank journal under the different conditions. These observations being recorded, would enable us to estimate approximately the difference in the amount of radiation, and of heat generated by pressure on the crank, between the two principles of working.

When these suggestions have been carried out we shall be nearer the solution of the problem.

An important feature in the Compound Engine, and one which contributes to its safety and the general favour with which it is beginning to be regarded, is the greater uniformity of pressure on the crank pin, which obviates much risk, and averts much disaster.

This feature has already been noticed in one of its effects,—the great and unequal initial pressure on the crank pin, and crank shaft journal in the Simple, as compared with the Compound Engine. The improvement in this respect in the Engine which we have just had under consideration has been marked and decisive. The bearings connected with the crank pin and journal are not subject now to heating and wearing to near the same degree as before. If less heat be thus generated, then certainly less force is wasted.

Now, for a given amount of power there must necessarily be a given average pressure on the crank pin through a revolution,—and this average pressure must be in the circular line of its motion. For a high degree of expansion, the projectile force of the steam will propel the piston and other reciprocating parts with too much violence against the crank when on the dead centre, and at such angle as cannot give the most useful effect. If the force at these points of the cranks rotation be greatly in excess of the average pressure on the crank pin, then some portion of the excess of force will inevitably be transmitted through the crank into the crank shaft and bearing, where it cannot have any productive effect. Some part of the force therefore has been wasted, or practically destroyed. If, in a Simple Engine with a high degree of expansion, the pressure, inertia, and momentum of the reciprocating parts could be perfectly balanced throughout the stroke, then such Engine would do just as well as the Compound, so far as the equalization of the force.

In the Compound Engine made according to the rule of proportion, the initial pressure of steam in each cylinder will be maintained so far, that at the point of cut-off the crank will be at an angle where the pressure will be more usefully exerted; and the pressure of steam not falling as rapidly from the point of cut-off, the extreme momentum which is necessary in the Simple Engine with a high degree of expansion, will not be required here. For a given initial pressure and a given amount of expansion, the point of cut-off in the Simple Engine, as compared with the compound, will be as one to the square root of the number of expansions. For instance, suppose, as in the illustration of compound cylinders and diagrams already given, that the number of expansions be 16. The square root of 16 will be 4; and as the point of cut-off for 16 expansions in a Simple Engine will be one-sixteenth, then the point of cut-off in each cylinder of the Compound Engine will be four-sixteenths;

so that the point of cut-off in the Simple Engine will be one-sixteenth, and in the compound one-fourth. So, if we take nine expansions, the ratios will be as one-ninth to three-ninths. It will be clearly evident that with the distribution of the pressure in the compound cylinders as we have here shown, it will be less difficult to equalize the pressure on the crank during every degree of its rotation.

There is a *law* in the distribution of pressure of the steam in Compound Engines, which has not yet been noticed, and which may be regarded as one of the causes of their success. In every Simple Engine where the cut-off is not later than half stroke, the terminal pressure is only half the pressure at the middle of the stroke,—neglecting here all loss by condensation, or gain by re-evaporation, and assuming the steam to be in the inverse ratio of pressure to volume. In the Compound Engine, arranged according to rule of proportion, where the cut-off is not later than half stroke, the pressure in the middle is three times the terminal pressure whatever may be the degree of expansion. This will require a few words of explanation to make it clearly understood by every reader.

By the amount of pressure in the cylinders of the Compound Engine as here stated, is meant that the available and effective pressures in both cylinders are reduced to one cylinder,—which must always be the low pressure one. If, for example, we refer again to the theoretical diagrams on page 232, where the ratios of the cylinders are 1 : 4, and of which details of pressures, &c., will be found in table on compounding, which will be given hereafter. The presures in the two cylinders being found, that of the high pressure cylinder must be divided by four, and the quotient added to that of the low pressure cylinder. This will give the pressure at any part of the stroke, which is to be compared with the corresponding point of stroke in the Simple Engine. If we take 16 expansions, in order to make a comparison, we shall have the advantage of the illustration on page 232 for reference, as also the table on compounding. Here then we find that both the simple cylinder and the low pressure cylinder of the compound give a terminal pressure of 7·5lbs. The simple cylinder then will have a pressure in the middle of 15lbs; but as we allow 2lbs back pressure, the effective pressure on the piston will be 13lbs. The Compound Engine will

have just the same pressure in the low pressure cylinder; and to this must be added the effective pressure in the high pressure cylinder at the middle of the stroke. The pressure here will be 120lbs initial pressure $\div$ 2 = 60lbs absolute in the middle; then 60lbs — 30lbs back pressure = 30lbs effective pressure on the piston. As the cylinder is one-fourth the area of the low pressure cylinder (the latter being of exactly the same area as the cylinder of the Simple Engine) we shall now get 30lbs $\div$ 4 = 7·5lbs, which being added to 13lbs = 20·5lbs, the effective pressure of steam in the middle of the stroke in the Compound Engine; whilst in the Simple Engine it is 13lbs, and therefore the middle pressure is 1·577 times greater in the former than in the latter. If we deduct 4lbs for back pressure, = 11lbs vacuum, then we shall have—

Simple Engine = 11lbs.
Compound Engine = 18·5lbs.

The middle pressure in the Compound Engine is here found to be 1·68 times greater than in the Simple Engine.*

This calculation of the aggregate effective pressure on the piston at the middle of the stroke in the compound, as compared with the Simple Engine, is based simply on the law of MARRIOTTE; but other laws come into operation which affect the result, as will be explained further on, and increase the advantage of the Compound Engine in this respect still more. For a given range of expansion, and a given terminal pressure, the middle pressure (absolute) of the Simple Engine will be *less* than double the terminal, whilst the Compound Engine will give *more* than three times the terminal pressure; in each case measured by the diagrams, and all other things being equal. The examination of a great number of diagrams has amply proved the correctness of this conclusion. The actual result practically obtained from a Compound Engine of correct proportions and good construction, will give not less than twice the effective pressure at the middle of the stroke, which will be obtained from a Simple Engine with the same terminal pressure of steam.

*With the ratio of expansion here under consideration (16), the highest aggregate pressure at half stroke would be obtained by having four cylinders, in the ratio of 1, 2, 4, and 8; and the steam in each cut-off at half stroke. The extreme complexity of such arrangement, however renders it doubtful whether any real advantage would result over the employment of two cylinders. The experiment is worth trying.

This will be independent of the greater efficiency of one principle than the other arising from the comparative permanent condensation which may occur. We will give two examples to illustrate the principle and degree of middle pressure in the Simple and Compound Engines.

For the first illustration let us take Diagram No. 46. The absolute terminal pressure here is 6·75lbs. The absolute pressure at the middle of the diagram is 12·5lbs,—the atmosphere being 15lbs at the time of taking this diagram. Now twice the terminal pressure would be 13·5lbs, so that here the actual middle pressure is one pound less, or 1·85 times the terminal.

For the second illustration we will take Diagrams 42 and 43. Here the absolute terminal pressure is 5lbs, and the middle pressure is 8·5lbs. (See Diagram No. 43). As this latter bears a smaller proportion to 5lbs than 12·5 does to 6·75 in the first example, some explanation may be desirable. If the reader has gone carefully through the descriptive analysis of these diagrams, he will remember that the induction valve of the low pressure cylinder is open—more or less—during about three-fourths of the stroke. This being so, it follows that the whole of the steam which is found at the termination of the stroke has not entered at the middle; hence the relatively low pressure here. The diagram from the high pressure cylinder (No. 42) shows an effective middle pressure of 25lbs. As the cylinders are in the ratio of 1 : 2·9, then $25 \div 2{\cdot}9 = 8{\cdot}62$lbs, which will be the effective pressure reduced to the area of the low pressure cylinder. The absolute middle pressure in the second cylinder being 8·5lbs, then add 8·62lbs, transferred from the first, and we obtain $8{\cdot}5 + 8{\cdot}62 = 17{\cdot}12$lbs, which is 3·7 times the terminal pressure. Now $3{\cdot}7 \div 1{\cdot}85 = 2{\cdot}0$, so that the compound gives exactly double the amount given by the Simple Engine.

But does this proportion represent the true value of the difference? So far we have taken the full absolute pressure at the middle of the stroke. Now it is evident that there must always be some back pressure, or defective vacuum. We will assume the back pressure to be equal in both cases for our present purpose; and there is no reason whatever that the same amount of vacuum should not be obtained in the case of Diagram No. 43 as in No. 46, with the same conditions present in the condenser; indeed, it would be easier of attainment. Let the back pressure then be called 2lbs,

equal to a vacuum of 18lbs, with the atmosphere at 15lbs. The result will then stand as follows :—

Diagram No. 46 = 12·5lbs – 2 = 10·5lbs.
Diagrams Nos. 42 and 43 = 17·12lbs – 2 = 15·12lbs.

Then—

Diagram No. 46 = 10·5lbs ÷ 6·75lbs = 1·555 times.
Diagrams Nos. 42 and 43 = 15·12lbs ÷ 5 = 3·024 times.

It follows then that the actual effective middle pressures are, in the Simple Engine 1·555 times the absolute terminal pressure, and in the Compound Engine 3·024 times. This does not give the true value. In order to arrive at the correct value, we must compare the effective middle pressures corrected to the same terminal pressures, which we will now proceed to do. As the absolute terminal pressure in Diagram No. 46 is 6·75lbs, and in No. 43 is 5lbs, then the absolute middle pressure in the latter must be increased in the ratio of 5 : 6·75.

Therefore as 5 : 17·12 :: 6·75 : 23·112.

The effective pressure we shall now obtain for the Compound Engine at the middle of the stroke will be 23·112lbs – 2 = 21·112lbs, whilst the Simple Engine gives 12·5 – 2 = 10·5, being in the ratio of 2·01 to 1:

As 21·112 : 10·5 :: 2·01 : 1·0.

The comparison will be still more exact by correcting the terminal pressure of Diagram 46 to that of Diagram 43. The proportion will then be as follows :—

As 6·75 : 12·5 :: 5·0 : 9·26.

Deducting now 2lbs back pressure, we shall have 7·26lbs as the middle pressure for Diagram No. 46.

Then as 15·12 : 7·26 :: 2·08 : 1·0.

These cases have been selected to illustrate the principle under consideration, because the Engines coincide more exactly in all essential points for making a true and reliable comparison than any *others available.* They are similar in having a high degree of

expansion; in being steam jacketed; in having every vital part in good working condition; in having the quantities of steam consumed carefully ascertained; and in being altogether trustworthy in all the details given concerning them. The Compound Engine has not been selected because it gives the most favourable comparison in illustration of the principle; as it does not. So far from this being the case, we may say, that we have a considerable number of diagrams from Compound Engines which would have presented a much more favourable comparison. Such cases, however, would not have coincided so nearly in all their conditions to the Simple Engine selected. Diagrams 42 and 43 would have given a more favourable result if the cylinders had been proportioned and arranged on our principle of compounding. If, in Diagram 43, the steam had been completely cut off at or before half stroke, the result would have been more favourable in this case, because a higher middle pressure would have obtained with the same terminal pressure.

Let us take two more examples for comparison; and they shall be from Engines which have already been described in considerable detail. Diagram No. 13 represents the Simple Engine, and Diagrams 47 and 48 represent the Compound Engine. And first, No. 13. Here the absolute pressure in the middle of the stroke is 9 + 15 = 24lbs,—the diagram being on the scale of 30lbs per inch. The terminal absolute pressure would be 14lbs if the exhaust valve did not open to reduce it. As there must necessarily be some back pressure, we will allow 2lbs as before. This would give 22lbs effective pressure in the middle of the stroke, being 1·57 times the absolute terminal pressure. Now see Diagrams 47 and 48 from the Compound Engine. These diagrams are good representatives of both ends of the cylinders from which they are taken—the two from the high pressure cylinder being almost as closely corresponding in every point as if they had been taken from one end only. Of the low pressure diagrams there is but the slightest difference between them. The mean middle pressure of the low pressure diagrams, represented by Diagram 48, is 3lbs + 15 = 18lbs, and the mean terminal pressure is 9·3lbs. Diagram 47 is taken to a scale of 35lbs per inch, and the effective middle pressure is 29·75lbs. As the cylinders are in the ratio of 1 : 3·1, then 29·75 ÷ 3·1 = 9·6lbs reduced to the area of the low pressure cylinder.

Then 18 + 9·6 = 27·6lbs, which will be the absolute pressure at the middle of the stroke on the area of the low pressure piston. The absolute terminal pressure being 9·3lbs, then 27·6 ÷ 9·3 = 2·9677 times. This result being slightly below the theoretical requirements as stated above, demands a further analysis. We need not remain long in a state of perplexity concerning this.

If the reader will turn once more to Diagrams 47 and 48, he will see that in the former the back pressure at the point corresponding to the highest initial pressure in No. 48 is 17·5lbs (unfortunately the decimal 0·5 has been omitted on diagram 47), whilst the highest pressure in 48 is 10lbs above the atmospheric line. Here then we see that considerable and unnecessary attenuation of the steam occurs. Instead of having a back pressure in No. 47 of 7·5lbs above the highest pressure in 48, there ought not to be more than 1·5lbs, and this would amply suffice if the steam could freely flow from one cylinder to the other. This would give in the middle of the high pressure cylinder 35·75lbs effective pressure on the piston; and when reduced to the area of low pressure cylinder would be 35·75 ÷ 3·1 = 11·2lbs. Then 11·2 + 18 = 29·2lbs, and when divided by 9·3lbs, which is the terminal pressure, will be 3·14 times.

By carrying the comparison a little further we shall be able to arrive at the practical results. Taking the pressure just found as the true factor from which to work out the correct conclusion, then 29·2lbs — 2lbs back pressure, in second cylinder, = 27·2lbs. The examination of Diagram No. 13 showed an effective middle pressure of 22lbs, with a terminal pressure of 14lbs. The true comparison between the two cases will be to increase the amount of pressure in No. 48 in the ratio of 9·3 : 14, which represent the terminal pressures of the Simple and the Compound Engines. Then as 9·3 : 29·2 :: 14 : 43·957, so that the Compound Engines will give—say, —44lbs, and the Simple Engine 24lbs absolute pressure at the middle of the stroke. Deducting as before, 2lbs back pressure, we shall have 22 and 42lbs as the effective pressures—being for the compound 1·91 times the amount of the Simple Engine.

This result is not quite correct, because the middle pressure of the compound has been increased in a corresponding ratio with the higher terminal pressure of the Simple Engine, whereas the middle pressure *of the latter* should be reduced in the ratio of the lower terminal

pressure of the former. As a deduction of 2lbs back pressure is made in every case to give the effective pressure, the proportion which such back pressure bears to the effective is as the amount of the latter. For a true comparison then we may present the following view :—

As 14 : 24 :: 9·3 : 15·943.

Then 15·943lbs — 2lbs = 13·943lbs effective middle pressure in the Simple Engine ; and 29·2lbs — 2lbs = 27·2lbs effective middle pressure in the Compound Engine.

As 13·943 : 27·2 :: 1·0 : 1·95.

Therefore the compound gives 1·95 times the effective middle pressure given by the Simple Engine.

The result here is not quite as good as in the first comparison. But these cylinders, be it remembered, are none of them steam jacketed. And besides, Diagrams 42 and 43, representing a greater number of expansions, will necessarily give a higher proportionate middle pressure relatively to the terminal pressure, all other things being equal; the inferior result therefore, of 6·2 expansions, is what we should naturally expect. A set of diagrams, which we have recently obtained from a pair of Compound Engines, in which the ratios of cylinders are as 1 : 5·6, and the initial absolute pressure is 150lbs, give the absolute middle pressure, reduced to the area of the low pressure cylinder, as 3·72 times the absolute terminal, which is 8·4lbs. The Engines are quite new (erected in the early part of 1872), and in good condition, so that the above result may be relied upon as correct and trustworthy. That no loss is taking place to reduce the terminal pressure unnecessarily, is proved by the fact that the actual terminal pressure corresponds exactly with the vario-thermal pressure law. The cylinders are not steam jacketed, or the result would probably be a little better. The diagrams are not included in the number given in this work, as they were not obtained early enough. The Engines belong to Mr. Charles Suthers, Oxford Mill, Oldham.

Many sets of diagrams from Compound Engines exhibit the practical operation still more favourably, but as the conditions of the Engines are not *thoroughly* reliable in all respects, we withhold them.

It might innocently be supposed that this advantage of the Compound Engine may be gained at the cost of some corresponding loss. That such is not the case is amply proved by the case of the Simple and Compound Engines, from which Diagrams 13, 47, and 48 were taken, and the results of which have been detailed in an earlier part of this chapter, so that any further reference here will be altogether unnecessary. So far from any loss resulting from the relatively lower terminal pressure in the Compound Engine, the facts which we have given with considerable circumstantiality elsewhere, prove beyond the possibility of doubt, a very great gain. The final and inevitable result at which we arrive, is that the principle which we have discovered and above developed, is a gain, not only without any corresponding loss, but one which is accompanied by other results equally important and beneficial. These are facts of the highest import to the engineering world.

As the question may arise in the minds of some readers why in the Simple Engine the middle pressure should be lower, and in the Compound Engine higher than the theoretical amount, relatively to the terminal pressure, it may be desirable that the question should receive a little consideration here, although it will be discussed more fully hereafter in another connection. For a given amount of expansion there is a much larger initial condensation in the Simple than in the Compound Engine, and a larger re-evaporation is the consequence, giving a larger amount of steam at the end of the stroke than at the middle. In the Compound Engine, where the cut-off is necessarily later, the initial condensation is small relatively to the measure of steam used, so that there is a smaller proportion available for re-evaporation. If no initial condensation occurred, then, even if all the steam subsequently condensed by its production of power should be re-evaporated, the terminal pressure would be on the vario-thermal line. This will be explained further on. The full value of the conclusion which we have now established, as to the proportionately greater middle pressure in the compound than in the Simple Engine, will be better apprehended when we have exhibited an important and interesting fact connected with the laws of Dynamics, associated with which, the real significance will become apparent.

It has already been stated, that in proportion to the speed of the

piston and the weight of the reciprocating parts, will be the pressure of steam required to give acceleration of motion; and that whatever force is required to effect this, exactly the same amount of force will be given out again by the reciprocating parts as their speed is retarded. The greatest acceleration occurs at the commencement and early part of the stroke, and decreases as the speed increases, until the piston has reached its highest velocity, when acceleration ceases, and retardation of speed begins.

At this point of highest velocity, which is at half the length of the stroke (taking the mean of the forward and back strokes), the forces of inertia and momentum—that is, of vis-inertia and vis-viva —are, for the instant, in abeyancé, neither being operative. The pressure then, on the crank pin, at this point of the stroke, is *exactly as the pressure of steam* on the piston or pistons (neglecting here the small amount of power absorbed by the friction of the piston and and slides, &c.), which, as we have just seen, is greatest in the Compound Engine. This we believe to be a true explanation of the cause, or, at least one of the causes, of the superiority of the Compound over the Simple Engine, where a high degree of expansion obtains; for though theoretically the sum of the rotative forces may be the same wherever the pressure may be exerted, yet by a more equable distribution, a more effective application of the pressure is secured, and a smaller proportion of energy dissipated.

It will be useful if we illustrate the subject, in order to enable the reader clearly to understand the operation of these laws, and the practical results in the working of the Steam Engine. We will take first the case of the Simple Engine represented by Diagram No. 13. The speed is 48 revolutions per minute, the length of stroke 5 feet, and the diameter of the cylinder 26 inches, giving an area of 530 inches. We will assume the weight of the piston and all the reciprocating parts at 2,600lbs. This weight will require an average pressure of 6·15lbs, and an initial pressure of double the amount, or ($6{\cdot}15 \times 2 =$) 12·3lbs, to overcome the inertia of rest, and to give the acceleration of speed here necessary. The pressure of 12·3lbs is required at the commencement only—the amount gradually and nearly uniformly diminishing to the middle of the stroke, where the equilibrium is attained. The retardation during the latter half of the stroke is corresponding and reciprocal. Here it is important to observe that this will be true only with a connecting rod of infinite

length. With a connecting rod of any practicable length there will be a difference in the speed of the piston at front and back (or bottom and top) in the inverse ratio of the length of such connecting rod; that is, the longer the connecting rod, and the smaller will be the difference. The piston moves with the greatest velocity at the end of the cylinder farthest from the crank in the forward and the return strokes equally. The amount of pressure required to overcome the inertia of rest, &c., as given above (6·15lbs), is calculated for the mean of both ends.

If we suppose for a moment that the speed, weight of piston, &c., are such as to require an average pressure of 5lbs, then 10lbs will be the initial and terminal pressures, and the following view of inertia will represent the intermediate pressures without the use of decimals.

−9·	−7·	−5·	−3·	−1·	+1·	+3·	+5·	+7·	+9·

The sum of all the figures, divided by the number of spaces, gives the average of 5lbs.

Having the Diagram No. 13 divided into 10 spaces of equal length, and having ascertained the steam pressure of each space, begin at the steam end, deduct from the first five spaces, and add to the latter five spaces, as in the following paradigm of diagram. This will give the pressure of steam exerted on the crank, minus the friction of the reciprocating parts, which is a variable quantity, and can only be determined by experiment at intervals. The spaces in the Indicator diagram being measured by the pressure scale, the amount of each is placed in the second line of the paradigm; the pressure of inertia having been found, is placed in the third line; the fourth line represents the incidence of pressure, or the actual pressure exerted on the crank at every part of the stroke. The first line in each paradigm is simply the numbering of the spaces.

1.—Paradigm of Diagram No. 13.

1	2	3	4	5	6	7	8	9	10
67·	62·	42·	30·5	23·5	19·5	17·	14·5	11·5	8·5
−11·15	8·65	−6·15	−3·65	−1·15	+1·15	+3·65	+6·15	+8·65	+11·15
55·85	53·35	35·85	26·85	22·35	20·65	20·65	20·65	20·15	19·65

We will now take the Compound Engine represented by Diagrams 47 and 48, and work out the corresponding results, so that

we shall be able to make a comparison. But first we will make a correction in the high pressure Diagram No. 47. We have already shown that the concave curve back pressure line is unnecessary, and is produced without any corresponding compensation,—being due simply to the insufficiency of the exhaust passages.

As the highest pressure reached in the low pressure cylinder is only 10lbs above the atmospheric line, then if we allow 11·5lbs for the back pressure line in the high pressure cylinder, it ought to be sufficient. Adopting this as the potential and proper back pressure line, we shall get the pressures as exhibited in the second line of the second paradigm; and correcting for inertia as before, the figures for which are given in the third line, the incidence of pressure will be found in the last line.

2.—Paradigm of Diagram No. 47.

1	2	3	4	5	6	7	8	9	10
46·	51·3	49·	43·	37·3	30·3	23·3	18·6	15·	11·6
−11·15	−8·65	−6·15	−3·65	−1·15	+1·15	+3·65	+6·15	+8·65	+11·15
34·85	42·65	42·85	39·35	36·15	31·45	26·95	24·75	23·65	22·75

The low pressure cylinder has the same length of stroke and number of revolutions. The area of the piston is 1,660 inches; and the weight of piston, &c., we will assume to be 3,300lbs. The steam pressure in each space of the diagram (No. 48), of inertia, and the incidence of pressures, will be represented in the paradigm following :—

3.—Paradigm of Diagram No. 48.

1	2	3	4	5	6	7	8	9	10
19·5	20·	18·	16·5	15	12·5	9·5	8·	6·5	4·
−4·5	−3·5	−2·5	−1·5	−0·5	+0·5	+1·5	+2·5	+3·5	+4·5
15·	16·5	15·5	15·	14·5	13·	11·	10·5	10·	8·5

It will at once be seen that the Compound Engine gives a far better distribution of the aggregate pressures.

The view just presented is not quite correct in any one of the cases given. The second line in each paradigm contains the exact measurement of the effective pressure as found by the method adopted for *ascertaining from* the diagram the amount of power.

This will not be correct for the solution of the problem under consideration. Refer again to Diagram No. 13. The measured pressure for the calculation of power in the first division is 67lbs, and this is the amount given in the first space in Paradigm No. 1.

Now it is obvious that when the steam is admitted into the cylinder against the piston, its force will be just as much as it exceeds the pressure of steam on the other side of the piston, and which constitutes the back pressure for the time being; because, if the pressure on the opposite side should be equal, then the piston would be in equilibrio, and whatever might be the pressure of the steam, it would not tend to produce motion. As the initial pressure in Diagram No. 13 is 65 + 15 = 80lbs absolute, then this is the pressure operating on the piston, minus the amount on the other side above the absolute vacuum line. Assuming the diagrams from both ends of the cylinder to be similar (which is the fact), then, as the back pressure line averages 10lbs below atmosphere, or 5lbs absolute, for the first division of the return stroke in Diagram No. 13, we shall now have 80 — 5 = 75lbs effective pressure on the piston.

The explanation here given will be a sufficient guide to the clear understanding of all other divisions in the diagram, and also for all other diagrams, and we will now proceed to present to view once more, in a more correct form, those which have already been given, —placing them in immediate succession, so that they may be seen at one view, and the more easily compared.

4.—Paradigm of Diagram No. 13 (2nd).

1	2	3	4	5	6	7	8	9	10
75·	63·	41·5	30·5	23·5	19·5	17·	15	10·	1·
−11·15	−8·65	−6·15	−3·65	−1·15	+1·15	+3·65	+6·15	+8·65	+11·15
63·85	54·35	35·35	26·85	22·35	20·65	20·65	21·15	18·65	12·15

5.—Paradigm of Diagram No. 47 (2nd).

1	2	3	4	5	6	7	8	9	10
54·5	53·5	50·5	44·7	38·7	30·	22·5	16·	13·	2·
−11·15	−8·65	−6·15	−3·65	−1·15	+1·15	+3·65	+6·15	+8·65	+11·15
43·35	44·85	44·35	41·05	37·55	31·15	26·15	22·15	21·65	13·15

6.—Paradigm of Diagram of No. 48 (2nd).

1	2	3	4	5	6	7	8	9	10
20	19·5	17·5	16·	15·	13·	10·5	8·	7·	3·
−4·5	−3·5	−2·5	−1·5	−0·5	+0·5	·1·5	+2·5	+3·5	+4·5
15·5	16·	15·	14·5	14·5	13·5	12·	10·5	10·5	7·5

The second and last lines in each paradigm are reciprocal,—the incidence of pressure merely being changed from one point to another in the aggregate energy communicated to the crank. A glance at the figures will suffice to show the better distribution of the aggregate energy in the Compound than in the Simple Engine. In Paradigm No. 4, the pressure on the crank pin is seen to be 63·85lbs per inch in the first division or space, and (22·35 + 20·65 ÷ 2 =) 21·5 at half stroke; the former being 2·97 times the amount of the latter. In Paradigm No. 5, the first space equals 1·262 times the amount at half stroke, which will be the mean of the fifth and sixth spaces. In Paradigm No. 6 the difference is 1·107 times. The ultimate incidence of pressure may be still more equalized by increasing the weight of the reciprocating parts, or the speed of the Engine.

But, it may be asked, cannot this be done with the Simple Engine as well as with the compound? We will see.

As a further example we will next take Diagram No. 19, and select for our purpose the end in which the cut-off is at one-twentieth of the stroke. As in Paradigms 4, 5, and 6, the figures given in the second line will represent the pressure above that which is found on the opposite side of the piston at the same instant. The speed of this Engine is 52½ revolutions per minute,—the stroke 4 feet, and the diameter of cylinder 24 inches, giving an area of 450 inches, say. The weight of the piston and all the reciprocating parts we will assume to be 2,400lbs. This will require 12·75lbs initial pressure to give the initial motion to the piston. The result will be as in Paradigm No. 7, on the next page.

In this paradigm the first and second spaces are subdivided in order to present a more minute view in the early part of the stroke, where the fall in pressure is so exceedingly rapid. The inequality here is still greater than in Paradigm No. 4. By doubling the weight of the reciprocating parts, the figures which are subtracted from the second line for the first half of the stroke, and added for the secon

7.—Paradigm of Diagram No. 19.

1		2		3	4	5	6	7	8	9	10
66·5	44·5	28·	22·	17·	12·	10·	8·5	7·5	7·	6·	5·
−12·13	−10·87	−9·62	−8·38	−6·3	−3·8	−1·3	+1·3	+3·8	+6·3	+9·	+11·5
54·37	33·63	18·38	13·62	10·7	8·2	8·7	9·8	11·3	13·3	15·	16·5

8.—Paradigm of Diagram No. 20.

1		2	3	4	5	6	7	8	9	10
56	39	31	24	21·	18·5	16·5	15·5	14·	12·	3·5
−35·5	−31·5	−26·2	−18·8	−11·4	−4·	+4·	+11·4	+18·8	+26·2	+33·5
20·5	7·5	4·8	5·2	9·6	14·5	20·5	26·9	32·8	38·2	37·

The spaces numbered consecutively on the first line of each paradigm are intended to represent equal units of length, and corresponding to the similar divisions in the Indicator Diagrams. It has been necessary to have these the full length of the page, because of the subdivision of some of the spaces.

half, will be doubled also. This would change the incidence of pressure on each side of the middle point, but here the pressure would remain exactly the same, so that the first half of the first division (being one-twentieth of the stroke) would then be 42·27lbs, and the tenth division would be 28lbs. The third division would give 4·4lbs, and the fourth 4·4lbs. No one will suppose that this would be a change for the better.

Take another example—the next one in order in the series of diagrams,—another Model Engine. (See Paradigm No. 8).

This Engine runs at a speed of 120 revolutions per minute; the stroke is 2 feet 6 inches, and the diameter 18 inches, = 250 in area—allowing a small deduction for the piston rod. In the above calculation we have assumed the weight of the reciprocating parts at 1,200lbs, which will require an initial pressure of 37·5lbs per inch to give the initial motion. In this paradigm the incidence of pressure in the last line is strangely distributed. If the weight of the reciprocating parts should be less than here assumed, then the figures in the last line would be changed proportionately.

The assumption of 1,200lbs as the weight of the reciprocating parts, is probably low enough, as they are made specially heavy for the dimensions of the Engine.

Above, it has been stated that the second and last lines in the paradigms, are equal in the aggregate.

In Paradigms Nos. 7 and 8 there seems to be a discrepancy. This is only apparent—not real. It arises by subdividing one or more of the spaces. The rectification is simple. Take the mean of the two subdivisions of the space in all the three lines, and the second and fourth line will then be found to be reciprocal. For example—in Paradigm No. 8, the mean of the first space is 47·5 in the second line, 33·5 in the third, and 14·0 in the last. These figures being substituted, will make the sums of the second and fourth lines equal. If there were 20 equal divisions, this correction would not require to be made.

It is quite evident now, that the forces cannot be balanced—nor nearly so—in the Simple Engine with a high degree of expansion. In an Engine producing a diagram such as No. 15, the balance of forces can be very approximately attained. Let it be assumed that the speed and weight of the reciprocating parts, &c., require 10lbs t produce the initial motion, then the following will be a corre representation.

This one will serve as a criterion in judging of the diagram from the low pressure cylinder in the illustration of theoretical compound diagrams on page 232.

9.—Paradigm of Diagram No. 15.

1	2	3	4	5	6	7	8	9	10
29·5	29·	26·	21·	17·	14·	12·	10·	8·	1·
−9·	−7·	−5·	−3·	−1·	+1·	+3·	+5·	+7·	+9·
20·5	22·	21·	18·	16·	15·	15·	15·	15·	10·

The high pressure cylinder there would be represented by the following:—

10.—Paradigm of High Pressure Theoretical Diagram.

1	2	3	4	5	6	7	8	9	10
90·	90·	85·	58·	38·	25·	17·	10·	5·	2·
−32·	−25·	−18·	−11·	−4·	+4·	+11·	+18·	+25·	+32·
58·	65·	67·	47·	34·	29·	28·	28·	30·	34·

The figures given as the pressures of inertia in the third line of Paradigm No. 10. by which to change the incidence of pressure, are not calculated from a known weight of piston, &c., but the latter must be made to the requirements of these or other figures which may be determined upon as the most suitable for distributing the incidence of pressure aimed at. The incidence of pressure given in space 10 can be diminished to any desirable extent by compression of the steam; and this should be done to such a degree as circumstances may render advisable,—such as to secure a smooth and even motion of the Engine.

Practically—as will be shown elsewhere,—the pressures in the second, or steam line, will be somewhat different from the amounts given in Paradigm No. 10. At one-quarter stroke the steam cannot be cut off instantaneously, as is represented in theoretical diagrams, so that the fall in pressure will be more gentle. Again, as the pressure is reduced, re-evaporation commences, tending to produce a higher pressure than the theoretical line. Furthermore, the steam contained in the fulcrum capacity, will still further increase the pressure from the point of cut-off to the end of the stroke. This *higher terminal* pressure will be advantageous, as it will compensate,

partially or wholly, for the fall in pressure which almost invariably takes place when the steam passes from the high to the low pressure cylinder.

In Paradigm No. 10 no compression has been taken into account. In order that it may be viewed as it will appear in actual working, when modified by the conditions just enumerated, we will represent it by the following:—

11.—Paradigm of Theoretical Diagram Corrected.

1	2	3	4	5	6	7	8	9	10
90·	90·	85·	64·	42·	29·	21·	13·	6·	−12·
−32·	−25·	−18·	−11·	−4·	+4·	+11·	+18·	+25·	+32·
58·	65·	67	53·	38·	33·	32·	31·	31·	20·

It will be altogether unnecessary to carry the investigation of this branch of the question any further here, as it will now be quite apparent that the range of expansion in one cylinder cannot advantageously be great,—say not more than four or five expansions; beyond the latter degree the inequality becomes very objectionable.

There is another branch of this subject, and though only a subordinate one, yet it is too important to be entirely neglected, but which so far has only been referred to provisionally, viz., the difference in the speed of piston between one end and the other of the cylinder. Now this feature is a source of disturbance, which in some cases may seriously affect the correctness of the conclusions arrived at in the manner of the foregoing paradigms. Therefore, as it is desirable to balance the forces as correctly as practicable, it will be advisable to effect the rectification by such regulation of the pressures as will accomplish this result.

In providing for inertia so far, it will have been observed that the figures representing the pressures of inertia in the third line of each paradigm are reciprocal in each of the corresponding divisions from the centre to the end. A simple method of arriving at the relative amount for each of the five divisions of the right and left hand halves of the paradigm, is to ascertain the mean initial pressure required for this purpose, and, having ruled a straight line on paper, and divided into 10 equal parts, measure with the pressure scale at one extremity of such line, and at right angles with it, the distance in pounds pressure required; then rule from this last point

a line intersecting the first one, or horizontal line, at half its length, and the distance apart of the two lines will be equal at both ends, and also at every other point equidistant from the middle, where the point of intersection occurs. The division lines for giving 10 spaces must be ruled at right angles with the horizontal line: and the pressure scale being applied at right angles also, in the middle of each space, will give the amount of inertia pressure to be placed there for addition or subtraction, according as it may be the place of retardation or acceleration.

As the angular vibration of the connecting rod generates a differential speed in the piston greater at one end of the cylinder than the other, a modification of the method just described will be needed to give the true incidence of pressure. It has been stated that the greatest velocity of the piston is at the end of the cylinder farthest from the crank. If the connecting rod be three times the length of the stroke, the highest velocity is attained when the crank is at 85·5 degrees from the centre line, and when the piston has reached the point 4·6 : 5·4 in the course of its stroke. Now the difference in the average or initial amount of force will be inversely as the difference of 85·5° and 94·5° at back and front of the cylinder. This difference obtains in the highest degree at the extremities. Whatever may be the amount of the mean initial pressure of inertia, the difference will be found by adding one-tenth of the amount at the end where the maximum velocity is acquired in the shortest distance, and subtracted equally at the opposite end. The decrease in the difference of pressure required as the distance increases equally from the two extremities is rapid, so that at 0·3 from each end the difference has been reversed, and the end of the cylinder, which, at the extremity requires the greater force to produce acceleration or retardation, at this point requires the less.

Notwithstanding such difference in the initial and terminal forces required for acceleration and retardation, yet the sum of the forces required is the same on each side of the point of highest velocity. The difference is in the distribution of such forces. Whatever force is required to impart a given speed to a given mass, the same amount of force must be given up again on being brought to a state of rest; the amount of energy in a given mass, at a given velocity, is a definite quantity; and whether the mass be brought to a state of *rest in* a longer or shorter space of time, it will yield the same

amount. No account is here taken of friction, because it does not enter into the operations here under consideration. Friction is a separate and distinct question.

With the piston of a Steam Engine, the force (or pressure) required to overcome the inertia of rest, and produce the acceleration of motion, is as the square of the angular velocity (number of revolutions), multiplied by the length of crank for any given weight of reciprocating parts. It is quite clear that the pressure per square inch will then be governed by the area of the piston. Let us suppose an Engine with an angular velocity of 50 per minute, a stroke of 5 feet—being a radius of crank of 2·5 feet,—a piston of 300 inches area, and weighing, together with all the reciprocating parts, 2,000lbs. Here the pressure of steam required to give the initial motion will be 18lbs; and will equal an average of 9lbs to give the necessary acceleration.

Now let us suppose that the same Engine is doubled in speed,—it will have twice the number of revolutions per minute, and its piston will pass through twice the distance in a given time, and it will have twice the maximum velocity. But the initial (and also average) pressure required will be more than double the former amount. The force required being as the square of the velocity, the initial pressure of inertia will now be $18 \times 4 = 72$lbs per inch.

Suppose now that we have an Engine of the same area and weight of piston (and all reciprocating parts), but with a stroke of 2 feet 6 inches, and an angular velocity of 100 per minute. In this case the number of feet passed through by the piston and the crank pin, per minute, would be exactly the same as with 5 feet stroke and 50 revolutions. The pressure required, however, would be twice as much—because the force necessary to overcome the inertia of rest, and impart the acceleration, being as the square of the angular velocity, multiplied by the length of the crank,—so, as the square of 100 is 4 times the square of 50, and the length of the stroke here is only half of the two former cases, the force required will be four times, divided by two, or twice the amount; or, taking it in pounds pressure per inch, it will be

$$\frac{18 \times 4}{2} = 36\text{lbs.}$$

The conclusion arrived at is as shown in the following cases:—

1st.—For a given number of revolutions per minute, a given length of stroke, a given area of piston, and a given weight of piston and reciprocating parts,—the force required for inertia will be ....1.

2nd.—For the same area and weight of piston, &c., but with half the length of stroke, and double the number of revolutions per minute—and therefore having the same maximum velocity of piston as in the first case,—the force required will be..2.

3rd.—For the same Engine as in the first case, with every factor remaining the same—except, that the number of revolutions shall be doubled,—the force required will be......4.

Above, we arrived at the conclusion that the amount of force necessary to generate a given velocity in a given mass, would be the same whether the time occupied in accomplishing it should be longer or shorter. The second of the three cases just given shows that the force required is twice the amount for the same velocity of piston, when the angular velocity of the fly wheel is doubled. Here is an apparent contradiction, which involves either a paradox or an absurdity. Let us endeavour to find the solution of the problem.

In connection with the paradigms of diagrams already given, it will be remembered that the amount to be deducted from the pressure in the steam line during the first half of the stroke, and which is absorbed by the vis-inertia of the piston, &c., up to its maximum velocity, or middle of the stroke,—and the vis-viva, or stored up energy of the piston, given up again during the second half of the stroke,—would be correctly (or nearly so) described by a straight line of a given length in 10 divisions, intersected exactly midway by another straight line, which would thus be equidistant at each end from the first, or horizontal line, and at such distance as the initial pressure required. Thus, one half would give the minus quantity, and the other half the plus quantity. Whatever may be the length of stroke, speed, weight, and area, the mean pressure will be described in this way—merely changing the amount of the initial pressure, thus described, according to the requirements of weight, speed, &c.,—all the spaces in the paradigm being measured by the same scale. The first and second cases given above, when worked out in this manner, *would still* show the initial forces, and also the sums of the forces, ***as 1 and 2.***

An important element, however, has yet to be introduced into the calculation.

Although the initial force required in the second case is double that of the first; and although the diagonal line (the hypotenuse) from the point of initial force, or pressure, will run straight, and intersect the horizontal line in the middle as before, and will therefore produce a triangle of double the area of the first,—if the same length on its base line; yet, as in the first case the stroke is twice the length of the second, so the base line of this triangle must be twice the length also; and thus, with only half the perpendicular height (initial force), but with twice the length, the area will be exactly the same. And so with the sum of the forces required to produce a given velocity in a given mass. As the velocity attained is the same, then, when more time has been occupied in attaining such velocity, the distance passed through must be greater. The simple solution of the problem is, that as in the first case there are twice the number of units of length passed through before the maximum velocity of the piston is attained, therefore, only half the force (pressure) is necessary for each unit of distance. Time and distance are equivalent in their relation to the velocity of piston as here defined, because, whatever greater length of time is occupied by the piston in acquiring a given velocity, a correspondingly greater distance must necessarily be passed through.

The principles here laid down will lead to the conclusion that, whether the distance be infinitely great or infinitesimally small, in which a given velocity of a given weight, may be imparted or arrested, it will absorb, and yield again, the same amount of force. This will be true in a large and comprehensive sense, but not in its application to the particular subject in hand—the transmission of all the motion imparted, without loss by transmutation into other forms of energy.

Such transmutation of energy as is here implied, will inevitably take place to some extent, if the arrangements be not such as to guard against it. It is therefore important that the subject should be well understood. Little, if any, loss need occur in a well arranged Compound Engine.

Not every reader will have clear ideas presented to him by some of the words used in treating this question, and therefore it may be desirable to define such words as they relate to the laws of motion, and especially in their application to the action of the piston,

&c., of the Steam Engine, so as to make the treatment as clear as possible.

Above, it has been stated that the amount of force required to overcome the vis-inertia of the piston, &c., will be as the square of the angular velocity. Now, the force here will be represented by a proportionate pressure of steam on the piston. But for this operation the steam must be supposed to exert its pressure during half the stroke only,—and during the acceleration of speed,—and when the maximum velocity has been attained, to cease to act. This force which will be of a given amount for each unit of the distance (that is, half stroke), must be conceived to have its initial pressure double its average pressure, and to diminish uniformly to the point of maximum velocity, where such pressure, or force, is supposed to cease. The resistance of the piston, to motion at first, and, at every successive point, to increase of velocity, is called its *vis-inertia.*

The piston having now acquired its maximum velocity, it receives no further force, nor will it give out again its accumulated energy unless its velocity be retarded. Science teaches us that a body in motion will continue eternally at the same velocity, if no resistance be encountered; and that a body at rest, will remain so eternally, if no force external to itself should act upon it to produce motion. This persistence of a body to continue in a state of rest, or a state of motion (of any velocity whatever) indifferently, is called *inertia.* The persistence of a body in motion when any resistance operates, is its energy, momentum, *vis-viva, or living force.* The amount of such inertia in the piston of the Steam Engine we shall have to consider and determine.

In the suppositious cases of Engines, given a little way back, the first and third cases give the difference in the force required by the vis-inertia of the piston, as the square of the velocity. Now, in these cases the comparative velocities are relatively the same when referred to the angular or to the rectilineal velocities. If we try to take a measure of the amount of force by the measure and pressure of steam required by the vis-inertia, we shall be able to arrive at a clearer conception of the nature and measure of the operation of the law.

Taking for granted just now the correctness of the calculated force required in case 1, let us ascertain why four times the *amount is* required in case 3. In the latter case we have *twice*

the speed; and, as the piston has only occupied *half* the time in attaining the maximum velocity, then twice the former amount of force would be consumed in imparting the same velocity (when imparted in half the time); but, as the maximum velocity is double what it was before, then the force requires to be doubled again. The amount of force per unit of length of the piston's motion, to the point of maximum velocity, must be measured, *first*, by the *time* occupied in acquiring any given velocity; and *second*, by the *velocity* attained. It is quite evident therefore, that as a double velocity is attained in half the time, a force of four times the amount for the interval of time during which it is acting, and for the distance over which it is acting, will be necessary.

Can we ascertain the amount of force consumed, in terms of the measure of steam required? Let us see.

Remembering what has been said concerning the true conception of the action of the steam in this connection, we may proceed at once to the computation of the theoretical consumption,—because, as already indicated, we are considering an operation which cannot obtain in practice, but still desirable to be conceived (as an abstract operation) in order to arrive at a true estimate of the measure of force. As steam constitutes the force applied to overcome the inertia of the piston, then, as four times the intensity of force is required, so four times the pressure of steam will be required, which will be four times the quantity for each stroke. But the speed has been doubled, and therefore twice the number of strokes will necessarily be made during the same interval of time, so that the fourfold quantity of steam has yet to be multiplied by two, making an eightfold consumption necessary for a twofold velocity. This may seem to be a very surprising result, and one which deserves some further investigation. That it is true need not be doubted.

Why is all this force required, and what becomes of it?

It is required, *first*, because the maximum velocity being attained in time 0·5, requires double the force which would be required in time 1·0; *second*, because double the velocity being attained in time 1·0, double the force is required; and *third*, the number of times which the maximum velocity is given, in a given interval of time, being doubled also, the consumption of force, compared with the first case, will be as

$$1 : 2 \times 2 \times 2 = 8,$$

during any given interval of time. Remembering that the vis-inertia and the vis-viva are reciprocal and equal, it is clear then, that whatever force is absorbed by the former, will be given up again by the latter; then, as the vis-inertia absorbs eight times the amount of force in case 3, which is required in case 1, so will the vis-viva give up again, in the form of motion (or otherwise), eight times the amount of force. If the speed should be tripled, then the force required per unit of length passed through by the piston, would be as $1 : 3 \times 3 = 9$; and the sum of the forces, as measured by the quantity of steam required during a given interval of time, would be as

$$1 : 3 \times 3 \times 3 = 27.$$

Coming now to the consideration of the application of the laws of motion to the difference of velocity, or rather the difference of *time* during which a given velocity is imparted, at the two ends of the cylinder, we shall be able to present the clearest views by illustrating in a series of paradigms. If we calculate for a connecting rod of three times the length of the stroke, it will be sufficient here, because this will serve as a guide in the calculation for any other proportion. The time occupied by the piston in acquiring the maximum velocity, at back and front, or bottom and top, is as $9 : 10$ nearly. The point of stroke at maximum velocity is $4{\cdot}6 : 5{\cdot}4$. In order to have the sums of the forces at each end equal, the point on the diagram of inertia, where the diagonal lines intersect, must be at $4{\cdot}5 : 5{\cdot}5$ of its length; and the depth representing the amount of pressure being inversely as the proportionate lengths, the sums of the forces at each end will be equal. Straight lines do not represent exactly the changes in the amount of force for every unit of length, but if drawn as just described, and as illustrated by Diagram of Inertia on page 270, they will be sufficiently accurate for calculations for the object under consideration.

For an illustration we will take diagram No. 13, so that when the result is presented, it can be compared with that given in Paradigm No. 4. We will give the same figures in the second, or steam pressure line; and in the third line, which represents the pressures of inertia, we will give the figures which represent the amount of the force in each space. The calculation shall be made for the same velocity, area, weight of piston, &c., as in Paradigm

No. 4. The maximum velocity of the piston is at the rate of 12·566 feet per second. As gravity would give a velocity of 10·062 feet per second, then as

$$10 \cdot 062 : 12 \cdot 566 :: 1 \cdot 0 : 1 \cdot 249;$$

therefore—

$$\frac{2600 \times 1 \cdot 249}{530} = 6 \cdot 127 \text{ lbs per inch},$$

which is the average of the pressure required for the mean of both ends. The amount must be doubled to give the initial and terminal pressures, so that these will be 6·127 × 2 = 12·254 lbs. This pressure will require to be increased at the end of the cylinder having the shortest distance to the point of maximum velocity (for connecting rod of three times the length of the stroke) in the ratio of 10 : 11, and decreased at the opposite end in the ratio of 10 : 9, whatever the amount of inertia pressure may be. The pressures will now be for the extremities,—left hand of paradigm, 13·479 lbs, and right hand 11·029 lbs. If a figure be drawn like the following diagram of inertia (page 270), the spaces can be measured by the pressure scale. Let the perpendicular height first be determined by the scale, and the hypotenuse at each end drawn to intersect the horizontal line at the point 4·5 : 5·5.

12.—Paradigm of Diagram No. 13 (3rd) Back.

1	2	3	4	5	6	7	8	9	10
75	63	41·5	30·5	23·5	19·5	17·	15·	10·	1·
−12·15	−9·15	−6·15	−3·15	−0·15	+2·15	+4·15	+6·15	+8·15	+10·15
62·85	53·85	35·35	27·35	23·35	21·65	21·15	21·15	18·15	11·15

13.—Paradigm of Diagram No. 13 (4th) Front.

10	9	8	7	6	5	4	3	2	1
1·	10·	15·	17·	19·5	23·5	30·5	41·5	63·	75
+12·15	+9·15	+6·15	+3·15	+0·15	−2·15	−4·15	−6·15	−8·15	−10·15
13·15	19·15	21·15	20·15	19·65	21·35	26·35	35·35	54·85	64·85

Paradigms Nos. 12 and 13 represent both ends of the cylinder; the first of them having the initial pressure of steam to the left hand,—being the end of shortest distance,—and the second having the initial pressure at the right hand.

The difference in the amount of inertia at the two ends will be more strikingly displayed if we present Diagram No. 20 in the same manner, adopting all the data given for Paradigm No. 8.

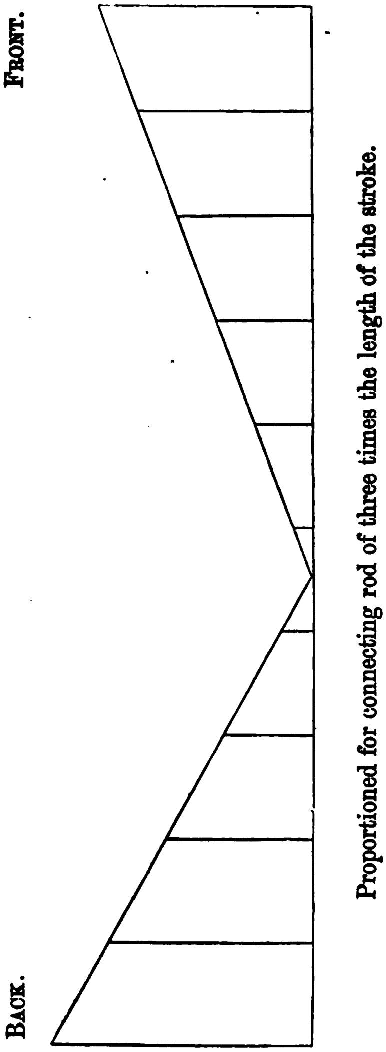


Diagram of Inertia (No. 50).

The incidence of pressure is here even more varied than the steam pressure. No. 14 shows the incidence of pressure in spaces two and *nine*, as 3·5 : 36·5, or as 1 : 10·4; while No. 15 shows the

14.—Paradigm of Diagram No. 20 (2nd) Back.

1		2	3	4	5	6	7	8	9	10	
56·	39·	31·	24·	21·0	18·5	16·5	15·5	14·	12·	7·	0·
−39·	−34·5	−27·5	−18·5	−9·25	−0·5	+6·5	+12·5	+18·5	+24·5	+28·875	+32·125
17·	4·5	3·5	5·5	11·75	18·	23.	28·0	32·5	36·5	35·875	32·125

15.—Paradigm of Diagram No. 20 (3rd) Front.

10		9	8	7	6	5	4	3	2	1	
0·	7·	12·	14·	15·5	16·5	18·5	21·	24·	31·	39·	56·
+39·	+34·5	+27·5	+18·5	+ 9·25	+0·5	−6·5	−12·5	−18·5	−24·5	−28·875	−32·125
39·	41·5	39·5	32·5	24·75	17·	12·	8·5	5·5	6·5	10·125	23·875

corresponding spaces as 6·5 : 39·5, or as 1 : 6·07. If the connecting rod should be shorter than here assumed, then the difference at the opposite ends would be still greater. In Direct-acting Engines the connecting rod very rarely exceeds three times the length of the stroke (which is the proportion adopted for these calculations), but is more frequently two-and-a-half times, and often still shorter,—sometimes even less than twice, as the writer has witnessed. When such is the case it is easy to understand the unsteady and even violent motion of the Engine when running at a high speed. A great amount of compression is necessary when so much energy is developed at the termination of the stroke as is seen in Paradigms 14 and 15.

The pressures of inertia, which are given in the third line of each paradigm, will be found by the following method:

Ascertain the maximum velocity in feet per second of the piston. Then ascertain the mean time required to attain such velocity. Next ascertain the velocity produced by gravity in the same time. Then, as the velocity produced by gravity is to the actual velocity of the piston, so must the weight of the piston and all the reciprocating parts be to the force required. This force, expressed in pounds, being divided by the area of the piston, will give the average pressure in pounds per inch on the piston; this amount being doubled, will give the initial and terminal pressures in pounds per inch, which represent the force.

In order to correct this mean pressure to the required difference at the two ends, for a connecting rod of three times the length of the stroke, increase the initial inertia pressure at the left hand in the ratio of 10 : 11, and decrease it at the opposite end in the ratio of 10 : 9. Construct a diagram of inertia similar to the one given on page 270. Mark by the pressure scale on the perpendicular line at each end, a point from which to draw the hypotenuses, each of which must join the horizontal line at 4·5 : 5·5; if the diagram of inertia be divided into 20 equal spaces, this point will be at 9 : 11. This latter number of divisions will be the better, as the units of length on each side the point of intersection will then be correctly proportioned,—giving 9 and 11. When these spaces are measured by the pressure scale, the sums of each end will be found equal.

The time occupied in acquiring the maximum velocity, as previously stated is as 9 : 10, but the intensity of the initial forces is

as 9 : 11. This arises from the fact, that at the end of shortest distance, the initial acceleration is still more rapid than the relatively shorter distance traversed, and that at the opposite end the acceleration is more nearly uniform during the first quarter of the stroke; and therefore, in order to represent the inertia nearly correct graphically, by straight lines, it is necessary to shorten the base and lengthen the perpendicular at the left hand, and lengthen the base and shorten the perpendicular at the right hand,—just so much that the hypotenuse at each end shall give the mean of the actual inertia, which can only be accurately described by curves. The real difference between the two extremities is somewhat greater than is shown by the diagram of inertia constructed on the rule here proposed.

This will be sufficiently correct, however, for all practical purposes, and will be found very convenient for use, as it can be worked out with the greatest facility, and with certainty.

If the connecting rod be two-and-a-half times the length of the stroke, the diagram of inertia will require the hypotenuse to intersect the base line at the point 4·25 : 5·75; or, if divided into 20 spaces, at 8·5 : 11·5; and the perpendicular lines must be changed from the mean initial pressure in the same proportion inversely.

The reader, whose experience may have been limited to the Beam Engine, or whose ideas may be associated with the Direct-acting Vertical Engine, will perhaps have wondered why the illustration of the subject should have been confined (or nearly so) to the Horizontal Engine. In order to represent the operation of the laws in the simplest manner possible, and divested of all unnecessary complication, the Horizontal Engine presents the best arrangement for such purpose. Here the whole mass of the reciprocating parts is free from the influence of gravity, so far as the operation of the laws of motion which have been under consideration. The parts are perfectly balanced, inasmuch as, when at rest, there is no tendency to move by their own weight. Whatever may be the weight of the reciprocating mass, the same amount of force will move it in either direction equally. To correctly illustrate the operation of the laws of motion, the reciprocating parts of the Horizontal Engine may properly be regarded as a weight suspended at the end of a cord of infinite length, and perfectly free to move without friction,—the whole mass moving the same distance, and therefore free from the

complication of having portions of the mass moving at different velocities.

The Direct-acting Vertical Engine, even when all its reciprocating parts move at the same velocity, does not meet the conditions for illustration so well as the Horizontal Engine, because the action of gravity is in the line of motion, and operating necessarily in one direction, thereby diminishing the pressure on the crank pin below the average pressure of the steam during the upward stroke, and adding thereto during the downward stroke, by the weight of the reciprocating parts. Equality in the incidence of pressure during the upward and downward strokes,—considering just now only the average, or aggregate, of each,—may be obtained by one or other of two distinct principles. The first, and certainly the best, would be to balance the weight of the piston, &c., by a beam having an equal weight at an equal arm, in the manner of the Beam Engine; or, if the free end of the beam should be shorter, which would be preferable, then the balance weight should be correspondingly increased. This arrangement would place the piston in equilibrio when not subjected to a pressure of steam. As the weight of the reciprocating mass will by this means be increased, so the sum of inertia will likewise be increased.

The best plan by which this principle could be carried into practice, would be to make the counter-arm of the beam of half or one-third the length of the other, as may seem best or most convenient, in any particular case, and attach the air pump to it. If this beam be made of wrought-iron plates, its weight will bear a very small proportion to the weight at the extremities. To whatever amount the air pump, and the weight of water lifted at each stroke, shall come short of that required to balance the piston, &c., it will be necessary to make up the deficiency by a dead weight on the extremity of the counter-arm of the beam. In the case of a pair of vertical cylinders working compound, the best plan probably would be as just described, but having two condensers, one attached to each beam; and this would give the advantage of withdrawing the water once for each stroke of the condensing cylinder. By having the cranks at right angles they would of course not lift at equal intervals, but this is a circumstance of only secondary importance.

Another principle on which the weight of the piston, &c., in the *Direct-acting* Ver[illegible] [illegible]ngine may be balanced, so that an equal

force at top and bottom shall be communicated to the crank, consists in giving a greater pressure of steam during the upward stroke,—such as the weight may require. To put the question in the concrete form, let us suppose the case of a Vertical Engine of four feet stroke, with a cylinder of 25¼ inches diameter = 500 in area, and having a piston, &c., of 2,000lbs weight. A very general arrangement of the Direct-acting Vertical Engine consists in having a guide beam, which is made to serve as the parallel motion for the piston rod, and to work the air pump in addition. These are so placed as to increase the weight of the reciprocating parts, and when, besides the weight of water lifted at each stroke, they are added to the piston, rod, and connecting rod, the total weight will then be more than just assumed, and all operating in one direction. Say the power requires a mean average pressure of steam of 12lbs per inch. The weight being 2,000lbs, and the area 500 inches, then the weight will be equivalent to a pressure of steam of 4lbs per square inch on the piston. Therefore, whatever may be the pressure of steam, 4lbs will require to be added for the downward stroke, and 4lbs. deducted for the upward stroke, to give the pressure exerted on the crank pin. The result will be correctly represented as follows:—

	Steam Pressure.		Equivalent of Piston, &c.		Pressure on Crank.
Top	12lbs	+	4lbs	=	16lbs
Bottom	12lbs	—	4lbs	=	8lbs

It is clear then that to give an equal pressure on the crank during each stroke, a rectification would require to be made thus:—

	Steam Pressure.		Equivalent of Piston, &c.		Pressure on Crank.
Top	8lbs.	+	4lbs.	=	12lbs.
Bottom	16lbs.	—	4lbs.	=	12lbs.

We are not forgetting the fact that another alternative is generally adopted, with a view of counteracting the effect of the gravity of the piston, &c., viz: the weighting of the fly wheel at a point opposite to the crank. This method is objectionable in a greater or less degree in every instance; and it is especially so when the speed of the Engine is considerable, as it generates a large amount of centrifugal force, which produces a very unsteady motion

of the fly wheel, and causes serious and dangerous vibration in all parts of the Engine. Such method of balancing may not produce any sensible amount of centrifugal force, and consequent vibration, when the framing, &c. of the Engine is *very strong*, and the speed *very slow*, as then it is little more than a balancing of ascending and descending weights.

As the amount of centrifugal force is as the square of the velocity, divided by the radius, for any given mass; and as such centrifugal force is equal at every degree of the circle; it follows then that the balance weight on the fly wheel must serve to counteract the effect of gravity in the reciprocating parts, and at the same time, to be truly efficacious, the reciprocating parts should, through the connecting rod, generate a centrifugal tendency in the opposite direction to that of the balance weight, and equal, at every degree of the circle. But this is not so, and cannot be. Only at, and near, the termination of the stroke, and therefore at 0° and 180° of the circle, can the motion of the piston, &c., counteract and neutralize the centrifugal tendency of the balance weight, even approximately. During the greatest part of the revolution such counteraction does not prevail. The centrifugal, or outward radial force, of the reciprocating mass, where too great, can be lessened to any desired extent, by compression of the steam, or where necessary, by the suitable admission of the steam before the termination of the stroke. It is quite clear that if a balance weight be required on the fly wheel of a Vertical Engine to counteract the centrifugal force of the reciprocating mass, and if such result be accomplished thereby, then it will be equally necessary in the case of the Horizontal Engine.

And where will be the counteracting tendency of the piston, &c., to the centrifugal force of a balance weight, when the effort on the crank is in the direction of the axis of the fly wheel? Beyond doubt, such a plan of balancing the weight of the reciprocating parts is altogether wrong in principle.

If we suppose the Engine last introduced for illustration to have a speed of 50 revolutions per minute, then the centrifugal force of a balance weight of 2,000lbs, at a radius of 2 feet, will equal 3406lbs. As a balance weight would be more conveniently placed at a greater radius than 2 feet, we will assume that it is placed at a radius of 8 feet. This being four times the radius of the crank, *then 2,000lbs* $\div$ 4 = 500lbs, which would at 8 feet radius be re-

quired to balance the weight of the piston, &c. The centrifugal force will be exactly the same as before. If the speed should be increased to 60 revolutions per minute, then the centrifugal force would amount to 4,904lbs; and at 40 revolutions per minute, it would only be equal to 2,180lbs. No one need be surprised at the dangerous amount of vibration and jarring which obtains in the working of many Direct-acting Vertical Engines, having balance weights, and running at a high speed.

The balance of weight will not be equivalent to the balance of forces, because the motions are different in kind. The weight which should properly be added to the fly wheel at the point opposite to the crank, must be simply to counteract and neutralize the centrifugal force which would be generated by the crank and connecting rod, if such provision were not made. If it were necessary to balance the weight of the reciprocating parts by a counter weight on the fly wheel, for the purpose of equalizing the centrifugal force, then it would be equally necessary to increase such weight as the pressure on the crank—and consequently as the pressure of steam—may be increased, because weight and pressure in this relation are exactly equivalent and coincident. What could be the difference in the effect on the rotation or steadiness of the fly wheel, between having, as in the case just supposed, an equal pressure of steam at top and bottom, with the gravity of the piston, &c., producing a double effective effort during one stroke over the other, and a horizontal, or any other balanced Engine, with the average steam pressure twice as much at one end as the other? Yet this latter is a condition which is frequently met with, and though an inequality in the rotative effort will certainly be the result, it will not be supposed that centrifugal force will be generated which will require a balance weight on the fly wheel to counteract and neutralize it, because, if so, such weight would only be necessary during one stroke, or half a revolution, which is a condition impossible of attainment. And if such balance weight be intended only to equalize the rotative effort on the crank during each stroke, by reason of the inequality of the pressure of steam, then, for the Horizontal Engine it must be placed at an angle of 90° with the crank, and its weight must be equal to the difference of pressure multiplied by the area of piston. If the effort of the piston on the crank generate centrifugal force, then the Horizontal Engine will require a balance

weight equally with the Vertical Engine, if such weight have the effect of counteracting the assumed centrifugal force. Moreover, the centrifugal force by such a principle, would increase with the increased effort on the crank, so that an increased pressure of steam would require an increase in the balance weight.

That the effort on the crank tends to move the axis of the fly wheel from its point of rest, or centre of motion, is undoubtedly true. This tendency can only be counteracted by having two cranks at opposite points, that is, at 0° and 180°, and as close together as possible in the line of the axis, with the efforts thus in the opposite directions, and equal, at any one instant of time. However irregular and unequal then might be the pressures and the efforts, if only simultaneous, it is perfectly clear that a balance weight on the fly wheel would not be either necessary or useful.

The contemplation of the principles just presented to view will lead to a clearer understanding of the cause of the unsatisfactory working of many Direct-acting Vertical Engines, and the disfavour with which, as a consequence, this class of Engines is regarded by many people. The Beam Engine being balanced, it has a close correspondence in the essential points just discussed, with the Horizontal Engine.

The calculations of inertia have been limited to the Horizontal Engine, in which the whole weight of the reciprocating parts passes through the same distance. The law is the same when applied to Beam, and other forms of Steam Engines, in which some portions of the reciprocating parts move at different velocities. In these cases it is necessary, before making the calculations in the manner of the foregoing paradigms, to reduce the inertia to the driving point; that is, the whole mass must be corrected for the velocity of the piston.

The above discussion of the questions of centrifugal force and the Direct-acting Vertical Engine, may seem to be an unnecessary digression from the main subject of this chapter; but being closely associated with one of the vital questions illustrating the superior advantages of the Compound Engine, it seemed useful, if not necessary, to treat the question of centrifugal force in connection with the Vertical Engine thus far. Besides, the questions are correlated in another way to one general, fundamental law. Whatever tends to disturb the free, smooth, and even motion of the various portions of *the Engine;* or creates unnecessary friction; or destroys the proper

balance of the forces, so as to generate conflicting efforts; all such conditions lead to a waste of power, by causing some additional portion of the energy of the steam to be transmuted into heat in the frictional parts of the Engine.

We will now give a table on Compounding according to our rule of proportion, with a full description following.

An examination of this table will be highly interesting and instructive. It is based on the law of proportion, and the calculations are made on the rule given on page 231, so that the reader may verify for himself any of the figures herein contained. All the data being given, there cannot be any difficulty in the matter. The table will be found of great advantage to all persons who may be contemplating the erection of Steam Engines, as it will enable them to see what dimensions of cylinders to adopt for any given pressure, and amount of power required.

The pressures of steam given in the table range from 35 to 145lbs above atmosphere; and, with the atmosphere added, which is absolutely necessary for all calculations, the initial absolute pressures are from 50 to 160lbs per square inch. The terminal pressure in every column is placed at 7·5lbs, or half on atmosphere, which we consider about the best as the basis of calculation.

The first line in the table gives the pressure of steam above atmosphere; the second line gives the atmosphere at 15lbs; the third line gives the absolute initial pressure. The fourth line gives the terminal pressure, which is the same under every degree of expansion. All these calculations, it should be observed, are based on the law of Marriotte, which is, that the pressure is as the inverse ratio of the volume. The fifth line gives the number of expansions, which means here, the number of volumes to which the initial volume is expanded. The sixth line gives the square root of the number of expansions. The seventh line gives the point of cut off in each cylinder, which, it will be seen by reference to rule, is determined by the number of expansions. The eighth line contains the hyperbolic logarithm, with 1 added, of the figure which represents the number of expansions in each cylinder of the Compound Engine.

The number of expansions in each cylinder of the Compound Engine is the square root of the total number of expansions. This hyperbolic logarithm number (and which has 1 added) when multi-

TABLE OF PROPORTIONS, &c. FOR COMPOUND ENGINES.

All calculations of Power in this Table are on the basis of 500 feet per minute speed of piston.	A	B	C	D	E	F	G	H	I	J	K	L
1 Pressure in pounds above atmosphere	35	45	55	65	75	85	95	105	115	125	135	145
2 Atmosphere or perfect vacuum	15	15	15	15	15	15	15	15	15	15	[illegible]	15
3 Total or absolute I.P. in pounds per square inch... ...	50	60	70	80	90	100	110	120	130	140	150	160
4 Terminal pressure in Condensing Cylinder	7·5	7·5	7·5	7·5	7·5	7·5	7·5	7.5	7.5	7·5	7.5	7·5
5 Number of expansions	6·666	8·	9·333	10·666	12·	13·333	14·666	16·	17·333	18·666	20·	21·333
6 Square root of number of expansions	2·583	2·828	3·054	3·266	3·464	3·653	3·830	4·	4·163	4·320	4·472	4·619
7 Point of cut-off in each Cylinder	0·3871	0·3534	0·3274	0·3061	0·2886	0·2737	0·2611	0·2500	0·2402	0·2314	0·2236	0·2165
8 Hyperbolic Logarithm of point of cut-off, + 1... ...	1·948	2·039	2·116	2·183	2·242	2·295	2·342	2·386	2·426	2·463	2·498	2·530
9 T.P. and B.P. 1st Cylinder, and I.P. 2nd Cylinder ...	19·357	21·216	22·92	24·5	25·98	27·39	28·72	30·	31·227	32·4	33·542	34·64
10 Average absolute pressure in High Pressure Cylinder	37·707	43·28	48·498	53·483	58·25	62·86	67·262	71·58	75·756	79·80	83·788	87·639
11 Average E.P. B.P. as per line 9 being deducted ...	18·350	22·064	25·578	28·983	32·27	35·47	38·542	41·58	44·529	47·4	50·246	52·999
12 Avg. E.P. in L.P. Cyl, 2lb deducted for B.P. (=13lb vcm)	12·61	13·3	13·87	14·372	14·815	15·212	15·565	15·895	16·195	16·472	16·735	16·975
13 Area in square inches of Low Pressure Cylinder	100·	100·	100·	100·	100·	100·	100·	100·	100·	100·	100·	100·
14 Area in square inches of High Pressure Cylinder ...	38·710	35·360	32·744	30·627	28·868	27·390	26·110	25·	24·021	23·148	22·361	21·647
15 H.P. of High Pressure Cylinder	10·762	11·810	12·690	13·450	14·112	14·701	15·247	15·750	16·206	16·624	17·020	17·352
16 H.P. of Low Pressure Cylinder	19·100	20·140	21·01	21·775	22·447	23·048	23·583	24·083	25·538	24·957	25·356	25·720
17 Total H.P. of both Cylinders	29·862	31·950	33·7	35·225	36·559	37·749	38·830	39·833	40·744	41·581	42·376	43·072
SINGLE CYLINDER.												
18 Point of Cut-off	0·1500	0·1250	0·1071	0·0938	0·0833	0·0750	0·0682	0·0625	0·0577	0·0535	0·0500	0·0468
19 Hyperbolic Logarithm, + 1	2·896	3·079	3·233	3·370	3·485	3·590	3·685	3·772	3·855	3·927	3·996	4·060
20 A. Ab. P. – 2lbs B.P. = A.E.P.	19·62	21·09	22·247	23·275	24·137	24·925	25·66	26·29	26·912	27·452	27·970	28·450
21 H.P.	29·878	31·958	33·707	35·265	36·572	37·765	38·844	39·833	40·775	41·594	42·378	43·106
22 With stroke at 60 in., Single Cyl. cut-off will be in inch.	9·	7·5	6·428	5·625	5·	4·5	4·09	3·75	3·46	3·214	3·	2·812
23 In each Cylinder of compound ditto	23·228	21·216	19·646	18·371	17·32	16·433	15·666	15·	14·412	13·88	13·416	12·99

plied by the terminal pressure in the cylinder which is being calculated, will give the average pressure throughout the stroke. Having by this logarithm obtained the average pressure, the back pressure must be deducted from it, and the remainder will be the average effective pressure on the piston. The amount of back pressure is easily determined. Whatever may be the terminal pressure in the high pressure cylinder, that also will be the amount of back pressure. The back pressure in the low pressure cylinder, is in every case taken at 2lbs,—being equal to a vacuum of 13lbs, which is near the best constantly attainable result.

The ninth line gives the terminal pressure in the high pressure cylinder, which is the back pressure also, and likewise the initial pressure in the low pressure cylinder. The tenth line gives the average absolute pressure in the high pressure cylinder,—that is, calculated from the line of perfect vacuum. The eleventh line gives the average effective pressure in the high pressure cylinder,—being the amount in line 10, with the amount in line 9 deducted. Line 12 gives the average effective pressure in the low pressure cylinder,—a back pressure of 2lbs (equalling a vacuum of 13lbs) having been deducted, so that the average absolute pressure would be 2lbs higher than the figures in each column. Line 13 gives the area in square inches of the low pressure cylinder, which in all calculations on the power of Compound Engines, must be taken as the basis. This area is fixed at 100 square inches under every pressure.

In all calculations on the comparative power of Simple and Compound Engines, it is essential to remember this:—The condensing cylinder must be of the same area and speed of piston for a given initial and a given terminal pressure (and therefore of an equal number of expansions) whether a single cylinder, or two cylinders of a Compound Engine be under consideration,—because the same quantity of steam will be consumed in one cylinder, in expanding to the same given terminal pressure, as if an additional cylinder had been used. The additional cylinder may seem to the uninitiated to give increased cylinder capacity, and therefore, if only the same quantity of steam be admitted, that it will expand to a lower terminal pressure in the low pressure cylinder. This, however, is not the case. It is necessary to bear in mind this fact: All the steam which exists in the high pressure cylinder at the termination of the stroke, must pass into the low pressure cylinder, which will

then contain all the steam that it would have contained had the steam been admitted direct from the boiler. The additional power gained by the high pressure cylinder, would have been gained equally, if the same steam had been admitted into the one cylinder of the Simple Engine, as was admitted into the high pressure cylinder of the Compound Engine,—so far as the calculated power of the steam. In practice the calculated power is not approximated to equally under different conditions.

Line 14 gives the area in square inches of the high pressure cylinder, which is to 100 (the area of low pressure cylinder) as the square root is to the number of expansions,—the latter being found in line 5, and the former in line 6. Line 15 gives the H.P. of the high pressure cylinder, line 16 of the low pressure cylinder, and line 17 gives the total H.P. of both cylinders of the Compound Engine. These powers are calculated on a piston speed of 500 feet per minute.

Line 18 gives the point of cut-off for a Single Cylinder Engine, with the same number of expansions, the same initial, and the same terminal pressures, as the Compound Engine. The number of expansions, of which the decimal numbers here are the reciprocals, will be found in the fifth line. Line 19 gives the hyperbolic logarithm, plus 1, corresponding to the point of cut-off, or number of expansions, which, being multiplied by the terminal pressure, gives the average. Line 20 gives the average effective pressure,—being the average absolute pressure, minus 2lbs back pressure. Line 21 gives the power of the Simple Engine, calculated on the same speed of piston as the Compound Engine, viz: 500 feet per minute.

Assuming the length of stroke at 60 inches, line 22 gives the distance traversed in inches by the piston of the Simple Engine at the point of cut-off; and line 23 gives the distance in inches traversed by each piston of the Compound Engine at the point of cut-off.

On examination of the figures in lines 17 and 21 of the table, it will be seen that the calculated power of the Compound and Simple Engine are alike. The slight difference which will be noticed in the two sets of figures is in the decimals only, and is due to the fact, that it is simply impossible to arrive at the same identical figures when working with interminable decimals.

We will now illustrate the table for practical application.

Suppose that it is intended to put down a new Compound

Engine, and that 500 H.P. be required. Say it is determined that the boilers shall be worked at a pressure of 85lbs per inch on the pressure gauge. Let there be a margin of 10lbs between the boiler pressure and the initial pressure in the cylinder. This will give us 75lbs + 15lbs = 90lbs initial pressure on the piston. Look under letter E in column, and the power will be found to be 36·5 H.P. for 100 in area of condensing cylinder. Then, as the power required is 500, divide this by 36·5, and multiply the quotient by 100, and the product will be 1370, the number of inches area required for the low pressure cylinder, which will be equal to a diameter of 41·75 inches. Now take the figures in line 6 and column E, and divide the area of the low pressure cylinder by them. Thus: 1,370 ÷ 3·464 = 395·5, which is the area required for the high pressure cylinder. Or, take this method, which will be more simple. As 500 H.P., divided by 36·5 H.P. = 13·7, so this latter must be multiplied by 100 to obtain the area of the low pressure cylinder, and by 28·868 (the number found in square 14 E) to obtain the area of the high pressure cylinder.

As it might be thought desirable to have a pair of Compound Engines for this amount of power, then each cylinder would require to be half the area just found.

We may illustrate in a more simple manner as follows: Look in square 17 E, and the power will be found to be 36·5 H.P. Now suppose that 365 H.P. be required, then the area of the low pressure cylinder will be 1,000 inches,—being 35·66 inches diameter; and the area of the high pressure cylinder will be 288·6 inches, and the diameter 19·3 inches.

It will be self-evident that 365 H.P. amounts to ten times the power given in square 17 E,—so the dimensions in 13 E and 14 E required to be multiplied by 10 also.

Take as another illustration column H in the table. Then, say 400 H.P. is required. Now, as 17 H shows 39·833 H.P., therefore the areas of the cylinders in 13 H and 14 H must be multiplied by 10 to give the power required,—the amount of power thus given being really 398·33 H.P. If it be desired to adopt a Simple Engine of one cylinder for the same power, then the same size of cylinder will be required as the low pressure of the Compound Engine.

There is another essential element in the question of compounding, and especially so with the steam cut off in each cylinder accord-

ing to our rule of proportion, and as worked out in the foregoing table, namely, a *receiver* between the high and low pressure cylinders. With Compound Engines arranged on our principle, a receiver is absolutely indispensable. It is required to equalize the back pressure on the high pressure piston, and the initial pressure on the low pressure piston. This receiver should be kept well covered, so as to prevent as much as possible the condensation of the steam within. Indeed, if the receiver, and also the cylinders were jacketed, and supplied with superheated steam, a considerable saving would be effected thereby. For a study on Steam Jacketing see chapter on that subject.

In the calculations of Compound Engines according to our rule of proportion, and as exhibited in the table, page 280, the initial pressure in the low pressure cylinder, and back pressure in the high pressure cylinder, exactly coincide. This is correct for theoretical diagrams, but in practice this exact coincidence will not obtain. There will almost invariably be a slight falling in the pressure of the steam in its passage from the high pressure cylinder to the receiver, and from the receiver to the low pressure cylinder. This fall of pressure will vary much according to the circumstances of each case.

If the cylinders be made with large ports, and with valves which open full wide, quickly, and at the proper time; if, also, the cylinders be steam jacketed at sides and ends, and supplied with steam of high temperature; if, furthermore, the receiver be surrounded by flues, or otherwise heated;—then the fall in pressure will be very slight. On the other hand, if the ports of the cylinders be small, or the valves do not open sufficiently, or not at the proper time; if the Engine runs at such speed as to render the valve orifices insufficient for the perfectly free passage of the steam; if the cylinders be not surrounded by steam (in jackets); then the fall in pressure between one cylinder and the other will be proportionate to the degree in which these defective conditions may exist.

When the steam issues from the high into the low pressure cylinder (or from the receiver), it comes in contact with surfaces which have just been exposed to the cooling influences of the condenser, and must therefore be condensed, and thus lowered in pressure, if external means be not provided for preventing this. Steam jacketing the cylinders, and superheating the steam in the receiver, *may be made* to fully counteract such tendency. It may however

be concluded that usually some fall of pressure will occur, as the necessary conditions to obviate this will not often be provided. Though such fall may be expected, its extent, as just pointed out, will depend on the construction. Some little compensation will be found in the terminal pressure in the high pressure cylinder, being above that due to the hyperbolic line from the point of cut-off.

This increased terminal pressure will arise from two causes;—first, the fulcrum capacity of the cylinder; and second, the initial condensation, and consequent re-evaporation, which will occur in the high pressure cylinder, whether it be steam jacketed or not. Thus a larger quantity of steam than shown by table will be supplied by the high to the low pressure cylinder, so that the initial pressure here may quite equal that given in the table.

There is a question of considerable interest, and which it is important to understand correctly; a question on which Engineers of high reputation fall into serious error, and therefore a question deserving of some attention. It will have been seen that we have treated the pressures in the two cylinders of the Compound Engine as though no space whatever existed between them which the steam must always fill. When we find educated and practical Engineers committing the mistake of including all the space between the two cylinders, in addition to the capacity of the low pressure cylinder, as the space which must be filled with steam at each stroke of the piston, and supplied by that which is exhausted from the high pressure cylinder, then it would not be wise to ignore the question, or pass it by with slight remark. Had it not been for the errors thus existing, we might have deemed it to be altogether superfluous to enter into any detailed examination and discussion of the principles involved.

If we suppose the cylinders to be in the ratio of 1 : 4, we can make use of the theoretical diagrams given in this chapter, on page 232, by which we shall arrive at a clearer conception of the principles and mode of operation.

By the proportions of the cylinders here adopted, the terminal pressure in the low pressure cylinder will be one-fourth the terminal pressure in the high pressure cylinder, if the fulcrum capacity in each cylinder bear the same proportion to its total capacity, and if the expansion follow the law of MARRIOTTE, or inverse ratio of pressure to volume. Further, the initial pressure will be inversely as

the capacity of the space at the point of cut-off in the low pressure cylinder is to the whole capacity of the high pressure cylinder,—less the loss by condensation or otherwise, as will be shown hereafter. The space between the two cylinders,—whether it may consist merely of a short length of pipes to convey the steam from one to the other, or of a capacious receiver equal to ten times the capacity of the large cylinder,—simply holds the steam stored up ready for use in the low pressure cylinder, when the valves shall open to receive it. Whatever may be the pressure in such receiver, the low pressure cylinder, if in proper condition, can only take in at each stroke a quantity corresponding with that discharged by the high pressure cylinder at the preceding stroke.

If it be supposed that the steam filling the low pressure cylinder and the space between the cylinders is discharged into the condenser at each stroke, then the pressure in the second cylinder will be as far below that due to the ratios, as the capacity of such waste space is to the capacity of the cylinders. This would certainly be a very wasteful and objectionable method of working a Compound Engine.

The space between the cylinders for the storage of steam, which we call the receiver, does not necessarily consume any steam. Whatever volume and pressure of steam issues from the high pressure cylinder into the receiver, will remain of the same volume, and pressure,—provided that there be no condensation here arising from loss of heat caused by conduction and radiation. A receiver serves the useful purpose of equalizing the pressure of steam between the high and low pressure cylinders; and just in proportion to its capacity, relatively to the capacities of the cylinders, and the point of cut-off in the low pressure one, will the amount of pressure in it approximate to a constant equilibrium. That such strange and erroneous ideas as pointed out above should come to be entertained and persistently maintained, by enlightened Engineers, is one of those phenomena which are sometimes difficult to understand. It is a phenomenon which may be fitly placed in the category of popular delusions.

That such a condition as the above idea implies, does sometimes exist, is beyond doubt, but that such condition is altogether unnecessary and injurious is equally certain. In the case of a Compound Engine in which the cylinders are some distance apart, and in which the induction valve of the low pressure cylinder is open the full

length of the stroke, and having its exhaust valve opening at three-fourths, or even earlier, as is sometimes practiced, then the loss of power here will be great in proportion to the area of the ports through which the steam may have to pass, the speed of the Engine, and the pressure of steam issuing from the high pressure cylinder. This would be simply making use of the low pressure cylinder as a thoroughfare from the high pressure cylinder to the condenser for a part of the steam, without being properly utilized.

This question of the coincidence or difference of the back pressure line in the first cylinder, and the steam line in the second to the point of cut-off, or initial pressure line, is well worthy of a little further consideration here. As a considerable difference is usually found to exist, it is believed by many to be invariable and inevitable, and is pointed to by those who are opposed to the principle of compounding as a proof of loss occurring in the Compound Engine which has no parallel in the Simple Engine. It is shown in another place that much of the difference of pressure, or apparent loss, is due to friction of the steam resulting from the insufficiency of the passages, and the great distance traversed. It has been shown how the loss arising from this source can be reduced to a minimum. Another source of loss, or reduction of pressure, is by the condensation of some portion of the steam on its admission to the low pressure cylinder. Let us examine the effect of this, and endeavour to discover whether the necessary result will be to create a difference between the initial pressure in the low, and the back pressure in the high pressure cylinders. Undoubtedly loss of steam will occur by initial condensation in the low pressure cylinder,—but does it necessarily follow that there will be a corresponding fall in pressure between the initial pressure here, and the immediate source of supply of steam.

Let us confine our attention just now to the Compound Engine on our principle, and having a receiver of such capacity that its contained steam, if not furnishing an unlimited supply, will approximate thereto, so far as the requirements of one stroke of the low pressure piston. The pressure in this receiver will thus correspond to an equal pressure in a boiler, and if a difference should obtain between this and the initial pressure in the cylinder, we know that it will be due to the insufficiency of the orifices through which the steam must pass. When the steam condenses on its first [illegible]

into the cylinder by contact with the internal surfaces, its disappearance will be instantaneously substituted by additional steam, which will maintain the equilibrium, if the valve orifices, &c., be made ample.

A practical proof of this is given in connection with Diagrams 45 and 46, on page 217, in which case the steam has to travel a considerable distance along the pipes from the boilers to the Engines. Where the reservoir of supply is close to the cylinder, the equilibrium will be more easily attained.

Suppose the capacity of the low pressure cylinder to the point of cut-off,—including fulcrum capacity,—is exactly equal to the measurement of the steam discharged from the high pressure cylinder at each stroke. Then, if no initial condensation takes place in the low pressure cylinder, and no fall in temperature occurs, the initial pressure here will coincide with the terminal pressure in the high pressure cylinder. But as a fall in temperature by the performance of work, and a greater or less amount of condensation by contact with the metal, will inevitably take place where special provision for superheating the steam before entering the low pressure cylinder has not been made, let us assume that 10 per cent. (it may be more) of the pressure available for supply, as discharged from the high pressure cylinder, has been lost, as shown by the reduced pressure in the second cylinder to the point of cut-off. Suppose the terminal pressure in the first cylinder to be 25lbs, then, with 10 per cent. initial condensation, the initial pressure in the second cylinder would be 22·5lbs absolute. But at what place may this fall be expected to occur? As already pointed out, if the valves be efficient, an equilibrium will be maintained between the initial pressure in the second cylinder and the receiver, because when condensation occurs on the admission of the steam, a further supply will make up for the loss sustained. This will have the effect of reducing the pressure in the receiver to near the equilibrium of the initial pressure in the low pressure cylinder, and therefore will permit the high pressure cylinder to exhaust on a line below its terminal pressure.

The fall in pressure which we have here indicated need not be regarded with feelings of astonishment or of dissatisfaction. Although there is not a corresponding fall in pressure in the course of expansion *in the Simple Engine*, yet, as we have abundantly proved, *the aggregate amount of condensation is far greater*. One result of the

fall in pressure in the manner just shown will be, that the high pressure cylinder will give a little more, and the low pressure cylinder a little less amount of power, than the proportions exhibited in the table on compounding, and so far there will be a closer approximation to equality, which will not be regarded with disfavour.

As it is necessary that a variation of power should be provided for, it will be advisable to point out how this can be accomplished in the Compound Engine which we have described. An examination of the table will show that with a terminal pressure of 7·5lbs, an amount of power will be obtained varying with the initial pressure and the number of expansions. Now it will be a very simple calculation to ascertain what will be the increase of power from a given increase of terminal pressure. Take for example column F in table, with 100lbs initial pressure, and 13·333 expansions,—yielding 37·75 H.P. If 7·5lbs terminal pressure gives 37·75 H.P., then 8·5lbs terminal pressure, it may be supposed, will give a proportionate increase of power. Here it is necessary to guard against error. If there were no back pressure in the low pressure cylinder,—that is, if in all cases the vacuum could be made perfect in the cylinder, then the terminal pressure would bear the same proportion to the amount of power in all cases, with the same given number of expansions. By increasing the terminal pressure from 7·5 to 8·5 in column F, the power will be increased from 37·75 to 43·2 H.P. The difference between 43·2 and 37·75, is 5·45 H.P. If 37·75 be divided by 7·5lbs terminal, the result is found to be 5·033 H.P. per lb terminal. It follows that one pound, or any other proportion, of additional terminal pressure, will give more than the proportion of power for each pound of the 7·5lbs in the table. And so for every pound below the terminal pressure of 7·5lbs, a deduction of 5·45 H.P. must be made.

It must be remembered now that in increasing or diminishing the terminal pressure for a variation of power, the increase of pressure will give a slightly greater, and the diminution, a slightly less, amount of power than the proportion of terminal pressure,—being in every case calculated according to the law of MARRIOTTE, and in every case having the same amount of back pressure or vacuum. That the higher terminal pressure should be found to give more economical results may at first sight seem surprising, and may seem

inconsistent with the conclusions arrived at elsewhere. Such, nevertheless, is the fact. The reader must not, however, rush to the conclusion, that a higher terminal pressure *simply*, is a gain. Other conditions must always be taken into account. In the suppositious case under consideration, though the terminal pressure is raised, yet the ratio of expansion remains the same, and the initial pressure is raised proportionately. The solution of the problem is simply this: That with any given ratio of expansion, the result is the best—all other things being equal—when the back pressure bears the smallest proportion to the average effective pressure. Economy, by the law of expansion—which alone we are considering here,—depending simply on the number of expansions, it follows, that if we have a high terminal pressure, we may, with a given ratio of expansion, secure a slight gain; but the gain would be still more by a higher ratio of expansion.

Returning to column F in the table, for illustration of variability of power, and assuming that we have a Compound Engine, the low pressure cylinder of which is 1000 inches in area, it is clear that the power at 7·5lbs terminal pressure will be $37{\cdot}75 \times 10 = 377{\cdot}5$ H.P. If the terminal pressure be increased to 8·5lbs, then the result will be 432 H.P. As, however, the number of expansions is 13·333, then the initial pressure will require to be increased by 13·33lbs. In reality, the increase in the initial pressure would require to be rather more than this, as will be fully explained further on.

If, as may be the case, the difference between the boiler pressure and the ordinary initial pressure in the cylinder, be not sufficient to allow the increased initial pressure in the cylinder necessary to give the additional power required, then it will be necessary to extend the point of cut-off in the high pressure cylinder only, and so maintain the true principle of expansion. If, on the other hand, a less amount of power be required, it may be done either by lowering the initial pressure, or by shortening the point of cut-off in the high pressure cylinder only.

Let us give one more illustration from the table, and show still more clearly the variableness in the amount of power which can be obtained. Take column H, with 120lbs initial pressure, and 16 expansions,—the cylinders being in the ratio of 1 : 4 in area, and the point of cut-off in each cylinder being at one-fourth the stroke. Now the table shows very nearly 40 H.P. for 100 inches area of the

low pressure cylinder. Then, if 400 H.P. be required, a low pressure cylinder of 1,000 inches area will meet the case.

Circumstances may occasionally arise which will necessitate the development of greater power than suffices for the usual requirements. If there be a margin of boiler pressure sufficient to allow the initial pressure in the cylinder to be increased to 130lbs, and the point of cut-off, in the high pressure cylinder only, be extended from one-fourth to five-sixteenths of the stroke, the power would be increased to 516 H.P. By reducing the initial pressure from 120lbs to 100lbs, and shortening the point of cut-off, in the high pressure cylinder only, to three-sixteenths of the stroke, the power obtained would then be 258 H.P., which is exactly half the amount just given above.

For the greater power (516 H.P.) the ratio of expansion is 12·8, and for the less, the ratio is 21·33. The terminal pressure in the former is 10·156lbs, and in the latter it is 4·6875lbs. The amount of power being as 2 : 1, it may be asked: What is the proportionate quantity of steam consumed? The answer will illustrate the question raised just above concerning the difference of economy resulting from the terminal pressure. The cut-off in the two cases are as 3 : 5, and the initial pressures are 100lbs and 130lbs. Then $100 \times 3 = 300$, and $130 \times 5 = 650$; therefore, whilst the amount of power is as 2 : 1, the quantity of steam consumed is as 2·1666 : 1·0, so that the lower terminal pressure is seen to give the best results. The absolute terminal pressures also show the comparative quantities of steam used, and will be found to bear exactly the same proportion. The same amount of back pressure has been allowed, and the same speed of piston assumed, in each case. Although the cut-off in the high pressure cylinder may be at three-sixteenths, the cut-off in the low pressure cylinder must still be at one-fourth, as given in the table.

If the initial pressure be still further reduced, say to 80lbs absolute, and the cut-off remain as in the last case, at three-sixteenths of the stroke, for the high pressure cylinder, the resulting pressures will yield—allowing the same back pressure or vacuum as before—200·38 H.P.

As the examples so far given assume a higher initial pressure than generally prevails, it may be well to give one having the initial pressure *much lower*. Take column B in table, where the absolute

initial pressure is 60lbs. With the boiler pressure at 60lbs by the pressure gauge, there will be a margin of 15lbs between this and the cylinder. Say the average amount of power required is 300 H.P. As the H.P. in column B is 31·95 for 100 inches area of low pressure cylinder, then, one of 1,000 inches area will give nearly 320 H.P. By raising the initial pressure to 70lbs (55lbs above atmosphere, still leaving a margin of 5lbs), and extending the point of cut-off from 0·3534 to 0·4 in the high pressure cylinder only, the Engine will then give 413 H.P. By reducing the initial pressure to 50lbs, and shortening the point of cut-off to 0·3 in the high pressure cylinder only, the amount will then be 230 H.P. In the former case the number of expansions is 7·07, and in the latter case, 9·425. The speed and back pressure are the same as before. Whilst the proportions of power obtained are as 1·8 : 1·0, the proportions of steam used are as 1·866 : 1·0. The examples now given show that *a considerable variation of power can be obtained by a very inconsiderable deviation from the positions fixed by the rule, and exhibited in the Table of Proportions for Compound Engines.*

Obviously, to accomplish these results, variable expansion valves will be requisite for the high pressure cylinder, and for the low pressure cylinder, valves which will have a definite and decisive cut-off will suffice. The cut-off in the high pressure cylinder may be made so as to be adjustable by hand, or so that it may be controlled by governors within certain limits.

It has already been incidentally observed that the cut-off in the low pressure cylinder should not be earlier than the point given in table according to rule. This condition is essential, because, if the cut-off should be earlier, the initial pressure in the low pressure cylinder would be higher than the terminal pressure in the high pressure cylinder.

In these calculations, as will have been noticed, no account has been taken of the clearance and the capacity of the ports, &c. When these are allowed for, it will be clearly seen that the high pressure cylinder will have a higher terminal pressure than the theoretical diagrams have shown. Referring back to the illustration given from column H, where the cut-off was assumed to be three-sixteenths of the stroke, it will be seen that when the capacity of the clearance, &c. (the whole fulcrum capacity) is added to the capacity of *the cylinder* produced by the motion of the piston to its point of cut-

l raise the terminal pressure proportionately, so that really amount of power will have been developed than the calcu- far have shown,—more especially if the cylinders be steam d.

the fulcrum capacity in the low pressure cylinder should unt to one-fourth the capacity of the cylinder produced by the ion of the piston to its point of cut-off at one-fourth the stroke, then the cut-off being shortened to three-sixteenths, the capacity ill just be equal to the capacity of the high pressure cylinder, exclusive of its fulcrum capacity. These considerations render it quite apparent that it cannot be advantageous to reduce the point of cut-off in the low pressure cylinder much short of the position determined by the rule of proportion, and given in line 7 of the table on compounding.

Nor must the cut-off in the low pressure cylinder be extended, because, if this have a larger capacity at the point of cut-off than the whole capacity of the high pressure cylinder, then a fall of pressure will occur in passing from the high to the low pressure cylinder, without performing any work, and this would be a departure from the true principle of expansion, and it would not give so high a coefficient of expansion. The effect of a fall in pressure from the high to the low pressure cylinder has been illustrated in the earlier part of this chapter, in treating of the variable amount of power obtained from a given quantity of steam in Compound Engines.

All the calculations and remarks so far on compounding according to our rule, have referred exclusively to those Engines having the two cylinders of the same length of stroke, and of the same speed of piston,—and having a rigid connection. It may be advisable to consider the application of the principle to Engines of other and different constructions.

In the Beam Engine McNaughted, the high pressure piston has only half the length of stroke of the low pressure piston; therefore the proportionate area of the high pressure cylinder would require to be twice that which is given in the table. In some cases a horizontal high pressure cylinder is connected with a Beam Engine, and beating at the same intervals of time,—being rigidly connected, —but the high pressure piston having a shorter stroke than the low pressure piston. In this case the area of the high pressure cylinder must be as much *greater* than the area given in the table as

the proportionately greater length of stroke of the low pressure piston.

There is another class of Compound Engines to which the application of this principle may be usefully considered. A considerable number of Beam Engines have been compounded, by placing at some distance,—usually in another room,—one, or a pair of, small horizontal high pressure cylinders. These are connected with the Beam Engines by gearing; and usually run two, and sometimes three revolutions, for one of the Beam Engines. To find the correct proportions of cylinders for this arrangement, it is only necessary to ascertain the speeds of the pistons in the high and low pressure cylinders. If the speed be the same in each, then proceed according to rule and table. If the piston of the low pressure cylinder should have a speed of, say 400 feet per minute; and the piston of the high pressure cylinder a speed of 500 feet per minute, then the area of the latter cylinder should be four-fifths of that given in the table.

A question of the highest interest and importance, in connection with the table on compounding, has been deferred until the end of this chapter had approached, in order that it might receive more attention than would have been suitable in the rapid sketch of the various features of the questions embraced. All the theoretical calculations so far made, have been on the basis of the law of MARRIOTTE, which gives the pressures as the inverse ratio of the volume of steam when expanding. This law is true for all dry elastic gases, when the change of volume is not accompanied by a change of temperature. Steam, when expanding under pressure, falls in temperature to a degree supposed to correspond to steam in the prescence of water from which it has been evaporated.

As the temperature falls with the falling pressure in performing work, the resulting pressure is lower than that due to the law of MARRIOTTE, because, in losing its heat it becomes contracted, and therefore is more attenuated than by the simple inverse ratio of pressure to volume. In the expansion of steam by heat there are two distinct laws. One is the law of GAY-LUSSAC, which gives *equal increments of volume for equal increments of temperature.* This is the law which applies to all dry elastic fluids,—steam being included when rendered dry by superheating, which can only be done when freed from the prescence of water. The other law of expansion of steam relates to the comparative volumes of steam at different

temperatures, in the prescence of water from which it is evaporated, and in which condition it is called "*saturated steam.*" In the first place, it may be said, roughly speaking, that the volume is in the inverse ratio of the pressure. In the second place, the volume increases by increase of temperature. Therefore, the volume of steam from a given weight of water, will be greater than the inverse ratio of volume to pressure: how much greater depends on the second law just stated. This expansion of volume cannot be expressed exactly by a co-efficient, as it is somewhat variable; that is, it is not uniform in its increase of volume by equal increments of temperature, through the practical range of steam pressure. An approximate co-efficient will be three-fourths of that of the law of GAY-LUSSAC for dry gas, which is 0·002035 increase of volume for each one degree Fahrenheit increase of temperature.

Instead of relying upon any co-efficient, it will be much simpler and clearer to turn to the table in appendix, on pressures, temperatures, and volumes of steam, established experimentally by the eminent French scientist, M. REGNAULT. It will be seen that as steam increases in pressure and temperature, its decrease in volume is less than the inverse ratio. At 15lbs pressure in the table, the volume of steam is found to be as 1,669 to 1 of water at its greatest density, which is 39·1° Fahrenheit. At 120lbs pressure the volume is 249. As the pressure of 15lbs is only one-eighth of 120lbs, it follows that the inverse ratio of volume to pressure would be $120 \div 15 = 8$ times the initial volume, which would give $1,669 \div 8 = 208·75$ volumes of steam to one of water at 120lbs pressure, instead of 249 volumes, as is actually the case,—being nearly in the proportion of 5 to 6.

This fact may be presented in a different form, and perhaps still more usefully. As the initial pressure of 120lbs equals 249 volumes; and as 15lbs equals one-eighth the pressure; then, by the law of inverse ratio, if the steam be expanded to 8 times the cubic capacity of the initial pressure, it will require the 249 volumes multiplying by 8, which will give 1,992 volumes, the corresponding pressure to which is 12·4lbs, and not 15lbs,—the difference being rather more than the ratio of 5 : 6. This will be the best and most reliable method of finding the real terminal pressure from any given initial pressure and number of expansions.

Returning again to the table on compounding, let us apply this

law, and process of calculation, to the pressures and capacities there given. In line 4 the terminal pressure is placed at 7·5lbs for every degree of expansion. This is correct when the initial pressure is divided by the number of expansions. When, however, we apply the table of M. Regnault, we arrive at a different result. Take column E in table on compounding. The initial pressure is 90lbs, the number of expansions 12, and the terminal pressure 7·5lbs. We find that at 90lbs pressure the steam has a volume of 323 to 1 of water, and at 7·5lbs it has a volume of 3180. If we multiply 323 by 12 we shall get 3876 as the volume of steam to one of water. This represents a pressure of 6·15lbs, and not 7·5lbs. Or, if we take 3,180 ÷ 12 = 265; then

265 : 323 :: 6·15lbs : 7·5lbs.

For a fuller illustration we will take column H in table on compounding, that we may have the advantage of referring to theoretical diagrams on page 232, corresponding with that column. Here the initial pressure is 120lbs, and the number of expansions 16. The volume of steam at 120lbs pressure being 249 to 1 of water, and which steam is expanded to 16 times its initial volume, then 249 × 16 = 3,984 volumes to 1 of water. By turning to table of pressures and volumes it will be seen that the latter number represents a pressure of 5·85lbs per square inch, whilst the terminal pressure by the law of Marriotte, or isothermal law, will be 7·5lbs, as seen in table on compounding, page 280. It is quite obvious from this view of the case, that a larger initial measure of steam will be required than is shown by the point of cut-off given in table, and determined by our rule of proportion, in order to obtain the full terminal pressure of 7·5lbs. The greater initial measure would require to be above that fixed by rule, in the proportion of 5·85lbs to 7·5lbs. But a greater average pressure would be secured by this means, if properly utilized, and not lost by excessive fulcrum capacity.

Practically, this increased measure of steam will exist without any special provision in most cases, because the fulcrum capacity in the high pressure cylinder (clearance, &c.) will be sufficient for this purpose. As the proportionate difference between the actual volume of steam and the inverse ratio is slightly greater as the number of *expansions* increases, so the proportion which the fulcrum capacity

bears to the initial measure produced by the motion of the piston to the point of cut-off, increases also, but in a much higher ratio. For example, we will take column H in the table on page 280, and consider the question as applied to the *Compound* Engine. Here the cut-off is at one-fourth; and assuming 60 inches as the length of the stroke, the cut-off will occur when the piston has traversed 15 inches. Now if the fulcrum capacity should amount to 3 inches sectional area of the cylinder (it is not well that it should be so much), this would be one-fifth of the measure shown by the Indicator Diagram, and would increase the actual initial measure, from what is apparent, in the proportion of 5 to 6. This would nearly compensate for the loss of terminal pressure pointed out above, which was from 7·5 to 5·85lbs, or in the proportion of 5 to 4 nearly, and would raise the terminal pressure to 7·02lbs, if no disturbing cause operated to affect the law under consideration.

Another cause will greatly assist in maintaining the terminal pressure, as given in the table; and still more in producing the average pressure there found. When the fulcrum capacity is added to the capacity produced by the motion of the piston to the point of cut-off, the fall in pressure will be less, and the expansion curve will consequently be so much higher. Moreover, the initial condensation which takes place, more or less, in every cylinder, will tend still further to raise the expansion curve,—giving to the high pressure cylinder certainly, a higher average pressure than that given in the table; so that although the terminal pressure in the second cylinder may be lower than that due to the isothermal law, yet the mean average effective pressure of both cylinders will be attained, if they be steam jacketed, and the valves good.

If we examine the Simple Engine with the same number of expansions (sixteen), we find that a fulcrum capacity as above, would be much too large for giving the terminal pressure in the table. As the cut-off in this case is at 3·75 inches traverse of the piston, then, even 1 inch fulcrum capacity would increase the volume of steam in the ratio of 1 : 1·266, or a little more than in the ratio of 4 to 5, which is the decrease of volume below inverse ratio. These calculations approximate very closely to the results arrived at by experiments with Compound Engines which are steam jacketed. When the cylinders are not steam jacketed, the fall in pressure will be greater than these *calculations* show.

It may be useful if we enter into a more detailed discussion of this question, and test the value of the above calculations by the results of actual working. Furthermore, let us examine the question in its connection with the Simple and the Compound Engine, by the aid of such data as may be relied upon. The actual terminal pressure, as compared with the initial measure and pressure of steam, is found to vary very much with the varying conditions which obtain. So many disturbing causes may come into operation in the action of the steam within the cylinders of Steam Engines, that it will be difficult, if not impossible, to predetermine the exact results, in any given case.

The analyses of a considerable number of examples of Simple and Compound Engines have led us to the conclusion that, for a given range of expansion, and a given initial measure and pressure of steam, there will be a much higher terminal pressure in the Simple than in the Compound Engine. This fact *alone* might point to either loss or gain as the result, and therefore it is of primary importance to ascertain the determining cause. If, for a given range of expansion, the initial condensation be large, then the re-evaporation will be sufficient to raise the terminal pressure above the hyperbolic, or isothermal line, because, when much condensation has taken place, a corresponding amount of heat has been transmitted to the metal, and this heat is given up again to the steam as it expands and falls in pressure. For a verification of this view, see analytical examination of Diagrams 45—46.

The initial condensation, relatively to the initial measure of steam used, and the pressure of steam found at the end of the stroke, is greater as the cut-off is earlier; by the diagrams just referred to, and others from the same Engines, we find the initial condensation, relatively to the terminal vario-thermal line, to be as follows:—

At 7·4	expansions	= 27·0	per cent.
„ 9·04	„	= 36·67	„
„ 11·4	„	= 46·67	„

These cylinders, be it remembered, are steam jacketed, and the diagrams from which the results just given are derived, were taken when the cylinders were thus heated. Turn now to Diagram No. 13. Here the cut-off is found to be at 0·1613, or 1 — 6·2, that is, when *corrected for the* fulcrum capacity, which is here taken into account,

and is assumed at two inches sectional area of the cylinder,—a quite sufficient and ample estimate for the valve arrangement in use. The initial condensation, when calculated by the same method, is 23·66 per cent. Now examine Diagram No. 22. In this, the number of expansions, when corrected for the fulcrum capacity, will be 11·5. The initial condensation here amounts to 57·8 per cent., unless the valves permitted the escape of steam into the cylinder after being shut. As the valves were quite new at the time,—having worked only a few months,—we need not suppose that leakage took place to any appreciable extent. These two last cases are not steam jacketed. Another case before us of an Engine with Corliss valves, and cylinder steam jacketed, with 9·8 expansions, shows 36·75 per cent. of initial condensation. Here the terminal pressure by the vario-thermal law—assuming the steam to be pure at the point of cut-off,—would only be 63·25 per cent. of the actual terminal pressure as shown by diagrams. The general result is, that, in the Simple Engine (single cylinder) the actual terminal pressure is higher than the calculated pressure,—being almost invariably higher even than the isothermal line, or inverse ratio of pressure to volume.

In Compound Engines, the actual terminal pressures are found to agree very closely with the vario-thermal law. By reading the analytical description of Diagrams 42 and 43, this will be seen and understood. This case is confirmed and strengthened by others similar. In Diagrams 47 and 48 the result is the same. By taking the full set of these diagrams, and fixing upon half-stroke in the high pressure cylinder, where we may fairly assume the steam to be completely cut off, the number of expansions will be 6·2 as in the case of Diagram No. 13. Here (in the Compound Engine, Diagrams 47 and 48), the actual terminal pressure is 5·6 per cent. higher than the vario-thermal line, and 9·6 per cent. below the isothermal line, or that due to the law of MARRIOTTE. In Diagram No. 13, the actual terminal pressure is 10 per cent. higher than the isothermal line, and 23·66 per cent. higher than the vario-thermal line. In an earlier part of this chapter, it has been shown by numerous facts which may be implicitly relied upon, that the Engine from which these diagrams (47 and 48) were taken, consumes only about two-thirds of the quantity of steam which was consumed before compounding, for the same power. It is evident from all the facts of this case, that the initial condensation of the steam is greater in the Simple Engine by

a considerable amount above what reappears by re-evaporation as the piston advances to the end of the stroke,—than in the Compound Engine.

In every Condensing Steam Engine, whether simple or compound, whether steam jacketed or not, when worked expansively, the steam, at the terminal pressure and volume, and immediately before being discharged into the condenser,—is charged with water in suspension, to a greater or less extent. This is the result of initial, and also of progressive, condensation. The quantity of water thus carried in suspension in the steam is evidently greater—all other conditions being equal—in the Simple than in the Compound Engine. The quantity of steam condensed, and thus carried away, is less when the cylinders are steam jacketed, whether the Engine be simple or compound. This fact is illustrated in the former class by the treatment of Diagrams 45 and 46, and in the latter class by the treatment of Diagrams 42 and 43.

The evidence furnished by the Simple and Compound Engines represented by Diagrams 13, 47 and 48, is indisputable and conclusive on the comparative amount of permanent condensation resulting from the two principles of utilizing the steam. It is true that the *absolute correctness* of the calculation, which we have given in the analysis of these diagrams in an earlier part of this chapter, *may be open to doubt*, and may be disputed, as one of the factors in the calculation is based on an assumption,—which is the evaporative efficiency of the boilers, and which, it will be remembered, we estimated at 7lbs of water per pound of coal. As the boilers are of the ordinary Lancashire type, and though well arranged and in good condition,—and having economisers (feed water heating apparatus) attached, yet without any specially superior arrangements and appliances for securing extra heating surface and consequent high efficiency,—this estimate may be considered just; and it is warranted by general experience. In connection with Diagram No. 46 it was stated that the evaporative efficiency of the boilers which supplied the steam under trial, was 6·96lbs of water per pound of coal,—and these boilers, we may observe, were constructed specially with the view of evaporative efficiency; but whether this object has been satisfactorily attained by their very unusual proportions,—being 24 feet long, 12 feet diameter, with two fire boxes of 5 feet diameter each,—must be left an open question.

By assuming the evaporative efficiency to be greater or less than we have estimated, the proportion of permanent condensation will be changed in both the Simple and the Compound Engines; and the ratio of such condensation, which may result from the two principles of working, will be changed likewise. But the vital fact still remains,—the greater permanent condensation in the Simple than in the Compound Engine. If we suppose the evaporative efficiency to be 6lbs only, then the superiority of the compound principle will be exhibited in a far higher degree. And if we suppose such efficiency to equal 8lbs of water per pound of coal, still the permanent condensation in the Compound Engine will only amount to 25·25 per cent., whilst in the Simple Engine it will amount to 49·25 per cent.,—giving a difference in the amount of permanent condensation, of exactly 24 per cent. of the whole amount of steam which we have estimated as passing into the cylinders for the generation of power. Remembering the proportion of permanent condensation as shown by the analysis of Diagram No. 46; and the fact of the greater proportion when steam is not in the jackets; and remembering, furthermore, that the difference will be—other conditions being equal,—greater as the range of expansion is greater; it is highly probable that the conclusion just arrived at, of 49·25 per cent. of permanent condensation in the Simple Engine, is near correct. The proportion of 24 per cent. less condensation on the whole consumption of steam, corresponds identically with the proportionate gain of the Compound over the Simple Engine, which we arrived at (page 242) by another and different process of calculation, and based on somewhat different data,—in connection with these same Engines.

Seeing that a correct estimate of the evaporative efficiency of the boilers in this case is of such unusual interest; and desiring to have the opinion of an impartial and well qualified authority on the question, we wrote to our friend Mr. Edward Ingham, Engineer, Oldham, asking him what he considered would be the evaporative duty of these boilers, and without indicating our own judgment,—he being well acquainted with the place, and the conditions necessary for forming a reliable opinion, and his knowledge and experience specially qualifying him for arriving at a correct estimate. His reply, dated September 16th, 1874, is as follows:—

"I should think that Messrs. ————* are evaporating 8lbs of

* Unfortunately we are not authorised to give the name and address of the owners.

water per pound of coal. They have good coal, a good economiser, patent wads in the flues, and every facility for a good result in point of evaporative duty. The boilers are cleaned once a month, and there is very little incrustation,—the water being all surface and domestic water, very soft, and impregnated with soap and other domestic refuse."

Considering the proportion of saving in fuel; and considering that every condition remained unaltered, except that of the change to the compound principle, the correctness of our calculations in regard to the relative and comparative condensation, based on this last estimate of the evaporative duty of the boilers, receives abundant confirmation, when all the known facts of the case are taken into account. No amount of ingenious reasoning, founded upon the law of expansion, or upon the greater area of surface—internal and external—of two cylinders than one, by reason of which, greater conduction and radiation may be assumed to result, can lessen the value of the facts of this case, or successfully assail the conclusion based upon them, confirmed, as such conclusion is, by a large number of other cases. The facts must be recognised, whether we can establish a theory which will satisfactorily account for them or not. In the chapter on steam jacketing some attempt will be made with a view to the solution of this problem.

Of the various causes which affect and modify the deportment of the steam within the cylinder, its *state* is one of considerable influence. When steam is superheated before entering the cylinder, its terminal pressure is found to be lower than when steam in its normal state of saturation is admitted. Careful experiments have also proved, that the terminal temperature is lower also; and that a less aggregate quantity of heat passes out of the cylinder into the condenser, as is evidenced by the fact of the less increment of temperature given to the condensing water. As superheated steam, although producing a lower terminal pressure, generates a greater amount of power for a given number of units of heat, so it may be concluded that, *the degree of saturation will be the measure of loss.* As the terminal pressure is lower when the steam is superheated, so probably it will be higher in the degree of its saturation, and the quantity of heat carried into the condenser will be greater for any given amount of power. This will indicate the importance of guarding against priming of the boiler, by which water in excessive quantities is carried into the cylinder.

Another conclusion which is drawn from the analyses of diagrams from Compound Engines *is noteworthy*. When the point of cut-off in the high pressure cylinder is late, then, for any given range of expansion after, the terminal pressure in the low pressure cylinder will be lower than when the cut-off is earlier for the same number of expansions. Of course this implies a difference in the ratios of the cylinders in the two cases. Where the high pressure cylinder is relatively small, then the cut-off, for a given number of expansions, must be later. The correctness of the conclusion just enunciated will be obvious on a full consideration of all the conditions involved. The later the point of cut-off in any given cylinder, and the nearer pure will be the steam,—all other conditions being equal.

In cases where a Compound Engine has only one of its cylinders steam jacketed, the terminal pressure will be above or below a given equilibrium, according as one or the other of them may be thus surrounded with steam. If the high pressure cylinder only be jacketed, then the initial condensation will be relatively less, and the re-evaporation more, in this cylinder; whilst in the low pressure cylinder there will necessarily be a larger initial condensation, and a relatively less re-evaporation. The result will be a lower terminal pressure in the low pressure cylinder relatively to the average pressure in both cylinders. If, on the other hand, the low pressure cylinder only should be jacketed, then the initial condensation in the high pressure cylinder will be greater, and the re-evaporation less; whilst in the low pressure cylinder, the steam, on its entrance, carrying with it a larger quantity of water in suspension, and coming in contact with metal of a higher temperature than in the former case, will be more copiously re-evaporated. Hence the terminal pressure will be raised. The result will be, that the terminal pressure in the low pressure cylinder will be higher relatively to the average pressure of both cylinders, than when the high pressure cylinder only is surrounded with steam in the jacket.

The Principle of the Co-efficient of Expansion of Indicator Diagrams, and the method of obtaining it, may now be advantageously considered, and its value determined. This principle of testing the efficiency of an Engine by the diagrams, will be open to serious objection if the proper precautions and corrections be not adopted. The co-efficient of expansion has hitherto been obtained by the following method:—Ascertain the average effective pressure,

If a Simple Engine, this will be easily done; if a Compound Engine, then, by getting the average effective pressure of each cylinder, and dividing the average of the high pressure by the number representing the ratio of the low to one of the high pressure cylinder, and adding the quotient to the average pressure of the low pressure cylinder. Having now got the actual average effective pressure, divide this by the actual absolute terminal pressure in the condensing cylinder, and the quotient will be the co-efficient of expansion.

This method would be correct if the terminal pressure in every case, in both Simple and Compound Engines, should be the same for any given initial pressure and volume, and range of expansion. We have seen that this is not so. Now, in order to ensure that the co-efficient shall be a true expression of the efficiency of the diagrams, it is evident that some method should be adopted which will produce one that may be relied upon as correct (or approximately so) for all cases. The following method will be found to give the correct co-efficient of expansion :—

Take the earliest point on the diagram at which the cut-off is seen to be complete. Then divide the whole length of the diagram by the distance traversed to such point of cut-off. The quotient will be the number of expansions, or the number of times to which the initial volume of steam will expand from such point. Now get the initial absolute pressure at such point of cut-off, and divide by the number of expansions; the quotient will be the theoretical terminal pressure by the law of MARRIOTTE. Ascertain now the actual absolute terminal pressure, as per diagram. This cannot always be done, and therefore requires a few words of explanation. As the exhaust valve almost invariably opens before the piston has arrived at the end of the stroke, the pressure rapidly lowers, and so would not correspond with the actual expansion line. Take the point immediately before the exhaust valve opens,—which is easily distinguished on the diagram,—and correct it to the terminal pressure. Thus, if the pressure at the end of the 9th division be 10lbs, then at the end of the 10th division it will be 9lbs. If the pressure be rising above the hyperbolic curve, the terminal pressure would, of course, be slightly greater, and *vice versa*. But this rule will be sufficiently near for the purpose in view. Now take the actual terminal pressure thus found, and the theoretical terminal pressure as found above; *add them together*, and divide by 2 to obtain the mean for a new

divisor. With this last as a mean terminal, divide the average effective pressure, as already shown, and *the true co-efficient of expansion* will be obtained.

A little attention to this subject will suffice to show the value of the method here described, for arriving at a true co-efficient of expansion for all cases. Turn to Diagram No. 22, where the cut-off, as per diagram, is at, say one-fifteenth of the stroke. The initial absolute pressure is 30lbs. The terminal absolute pressure is 5lbs,—being 2·5 times the amount given by the law of inverse ratio. Then, $5 + 2 = 7 \div 2 = 3{\cdot}5$lbs, which will be a divisor by which to obtain the true co-efficient. Now turn to Diagram No. 46. The cut-off, as per diagram, is at one-thirteenth of the stroke. The initial absolute pressure is 50lbs. Then 50lbs $\div$ 13 = 3·846lbs, which is the theoretical terminal pressure by the law of MARRIOTTE. Now the actual absolute terminal pressure is 6·75lbs,—or rather, this would be the terminal pressure if the expansion line should be continued to the end of the stroke, without being disturbed by the opening of the exhaust valve. Then 6·75lbs + 3·846lbs = 10·596lbs $\div$ 2 = 5·298lbs, which will be the divisor by which to obtain the co-efficient of expansion. Let us take an example where the terminal pressure falls below the isothermal curve. If we refer to Diagrams 42 and 43, the cut-off in the high pressure cylinder is at 1—6·5 of the stroke. As the ratios of the cylinders are as 1 : 2·9, then $2{\cdot}9 \times 6{\cdot}5 =$ 18·85 expansions. As the initial absolute pressure is 85lbs, then 85lbs $\div$ 18·85 = 4·509lbs, which will be the theoretical terminal pressure by the law of MARRIOTTE. The actual terminal pressure in this case is 5lbs absolute. The equilibrium will be $4{\cdot}509 + 5 = 9{\cdot}509 \div 2 = 4{\cdot}754$lbs. We are considering here only the cut-off as shown by the diagrams. When the fulcrum capacity of the high pressure cylinder is taken into account, the theoretical terminal pressure is found to be at 6lbs by the law of MARRIOTTE. See the analysis of Diagrams 42 and 43.

We will now examine Diagrams 47 and 48, from a Compound Engine already fully described. For the initial pressure and measure of steam, take half-stroke in No. 47. The cylinders being in the ratio of 1 : 3·1, the expansion from the above point will be 6·2. The initial pressure at half-stroke,—taking the mean of both ends,—is 64lbs absolute. Then $64 \div 6{\cdot}2 = 10{\cdot}3$lbs. The actual terminal pressure is 9·3lbs. The mean of these will be found to be 9·8lbs

The high pressure cylinder equals 30lbs, and the low pressure cylinder equals 13lbs, average effective pressure. Then, as the cylinders are in the ratio of 1 : 3·1, therefore, 30lbs ÷ 3·1 = 9·68lbs; and when added to the average pressure in the low pressure cylinder, the total effective pressure will be, 9·68 + 13 = 22·68lbs for the area of the low pressure cylinder. Then, 22·68lbs ÷ 9·8lbs = 2·314, which is the co-efficient of expansion. The principle of the co-efficient of expansion, and the correct method of obtaining it, will now be sufficiently clear.

Another principle exhibited in the Table on Compounding, merits some attention. It will be noticed that with the same terminal pressure of steam in each column (and therefore the same quantity), the power varies from 29·878 to 43·106 H.P.; whilst the initial absolute pressure is 50lbs in the former, and 160lbs in the latter case. Above, we have shown that to produce the same terminal pressure, a greater initial measure will be required than is shown in the Table on Compounding, by the proportion of the volume of steam at the reduced temperature of the terminal pressure below the isothermal law. If therefore, we take into account such increase in the initial measure, the average pressure will be raised accordingly, so that for a volume of steam evaporated from a given weight of water, a slight increase of power above that given in the table will be developed,—provided that the fulcrum capacity be small, so as not to absorb and render non-productive the whole of such increase of volume.

Were it possible to construct Engines entirely without fulcrum capacity, and the cylinders of a material perfectly non-conducting, then the terminal pressure would be still lower than we have yet arrived at. Under the conditions here supposed, the steam, when condensed by the energy which it expends, would not be re-evaporated. The reduction of the terminal pressure, on the expansion, &c., given in column H of the Table on Compounding, already stated, as being from 7·5lbs to 5·85lbs, would therefore be still further reduced by the proportion of the steam which may have been condensed by work done above the equivalent of the falling temperatures of the steam to the degree corresponding to the pressure. This proportion for the initial pressure and range of expansion found in column H of the table, page 280, would herein be 0·1953 of the whole quantity, so that the *terminal pressure, which* we have fixed at 7·5lbs, would be reduced by the

falling temperature, or variothermal law, to 5·85lbs, and by condensation resulting from the law of thermodynamics, to 4·71lbs. This result, for reasons given elsewhere, can never obtain in practice. The reader must clearly understand that this view has reference only to expansion against resistance, or the performance of work. In the above calculation of the terminal pressure by the thermodynamic law, the capacity of the water which results from condensation is disregarded, because, in comparison with the steam, it is so infinitesimal, and could not affect the conclusion arrived at,—being only as 1 : 5,000.

The three different terminal pressures here indicated are governed by different laws, which may be designated thus:—

1ST.—*Isothermal Law,*—or law of equal temperature,—being the law of inverse ratio of pressure to volume, and commonly described as the law of BOYLE and MARRIOTTE;—and producing by expansion, the hyperbolic curve, on which is based the hyperbolic logarithm.

2ND.—*Variothermal Law,*—being the law which gives the pressure due to the temperature of saturated steam corresponding to its increased volume.

3RD.—*Thermodynamic Law,*—which gives the line of still lower pressure than the variothermal law, by reason of a portion of the steam being condensed in the production of motion, or the development of power.

If superheating be adopted, then a still greater amount of power will be obtained from steam produced from a given weight of water, because, as it contains a greater amount of heat, it will have a slightly increased initial volume; and, what will be still more efficacious, its excess of heat will maintain the pressure at a higher average for a given weight of steam. It would not be advisable to superheat the steam *much,* before its admission to the high pressure cylinder, when the pressure there is high, as it would be destructive to the rubbing surfaces of the valves, piston, &c. A second superheating might be given to the steam, however, when in the receiver between the high and low pressure cylinders, which would be productive of good results.

There is one more fact in connection with the Table on Compounding, which is important and noteworthy. *For every degree of expansion, with cylinders arranged according to our rule, the power*

of the two cylinders will be exactly equal,—if the terminal pressure in the low pressure cylinder be coincident with the back pressure there. As in the table the terminal pressure is 7·5lbs in every case, so if 7·5lbs be allowed for back pressure in the low pressure cylinder, the two cylinders will give equal power. This is true only for the hyperbolic curve. If the initial pressure should be, say, half the amount given in the Table for any degree of expansion, then, with the correspondingly reduced terminal pressure, it is clear that the power of the two cylinders would closely approximate, even with the best vacuum attainable.

By the facts and reasons previously advanced, it may be concluded that such deviation from the proportion of power given to each cylinder in the table on compounding, as may be expected to occur, will be in the closer approximation to equality of power between the two cylinders. As the expansion line in the first cylinder may be expected to be higher than the isothermal line, it will give a higher average pressure; and as the expansion line in the second cylinder will certainly be lower than that given in the table, the average pressure will be less. If it be desired to give perfect equality of power by the high and low pressure cylinders, it can be easily done by adjustment of the cut-off,—usually by *extending* the point of cut-off in the low pressure cylinder.

Another principle should here be noted. The increase of power by increasing the number of expansions, has been exhibited in the table. *By squaring the number of expansions, double the amount of power will be obtained from a given weight of steam,—provided that in each case, the terminal pressure in the low pressure cylinder be coincident with the back pressure there,—and provided, further, that there be no condensation except that due to work performed, or that the condensation bear the same proportion to the amount of power in each case.* This law applies equally to the Simple and the Compound Engine.

This conclusion is deduced from the law of inverse proportion, or law of MARRIOTTE, which gives the pressure of steam as the inverse ratio of volume. As we have seen, this is not so exactly,—the higher pressure giving a volume larger than that due to the inverse ratio. It might be supposed that this law (the vario-thermal law) would interfere with the correctness of the conclusion just enunciated, and give a higher than double power, by squaring the number of *expansions.* But it must be borne in mind that the law prevails

throughout the whole range of practicable pressures; and although not with absolute uniformity, yet with sufficient approximation thereto, that the proportions are not sensibly disturbed by thus squaring the number of expansions.

It will be a very interesting calculation to ascertain (even though it may be only approximately) the weight of steam which passes through the cylinders in a given interval of time, or number of revolutions; and also what weight of steam may be consumed per I.H.P. per hour. If that steam only, which is found at the end of the stroke, as revealed by the pressure given by the Steam Engine Indicator, be taken into account, then the calculation will be extremely simple. But as steam is permanently condensed in a greater or less proportion in almost every case, it will be necessary to determine, as near as can be done, by such data as may be available, what the proportion may be. Absolute certainty can only be arrived at where the weight of water supplied to the boiler is accurately ascertained, and where all the steam evaporated from such water is supplied to the Engine. This view has been presented in connection with Diagram 46, representing a Simple Engine. Where this method is not available, it will be necessary to compute the proportion of permanent condensation according to the conditions of each case.

It will suffice here if we give one illustration from a Compound Engine: the one represented by Diagrams 42 and 43, and already treated at considerable length. By turning to the description of these diagrams, it will be seen that the absolute terminal pressure is 5lbs. The low pressure cylinder is 900 inches area; and the length, including fulcrum capacity, is 62·785 inches, which will equal 32·7 cubic feet per stroke. As the speed is 34 revolutions per minute, then $32{\cdot}7 \times 68 \times 60 = 133{,}416$ cubic feet of steam per hour, at 5lbs pressure. On examination of the table in Appendix, this quantity will be found to have a weight of 1,801lbs. The I.H.P. being 165, then, $1{,}801 \div 165 = 10{\cdot}915$lbs of steam per I.H.P. per hour.

Guided by the clearly ascertained results of other cases, we shall be enabled to arrive at an approximately accurate estimate of the proportion of permanent condensation, or the amount of water held in suspension in the steam at the termination of the stroke; and thereby determine the total weight of steam passing through the cylinders during a given interval of time. In connection with

Diagram 46, it has been shown that the initial condensation is greater as the cut-off is earlier; and the proportions for three different ranges of expansion are given on page 298. With 11·4 expansions the proportion of steam permanently condensed in the cylinder, and held in suspension in the steam at the termination of the stroke, was found to be 33 per cent.: see page 217. The range of temperature to which the cylinder would be subject, would be 175° Fahrenheit. Turning again to Diagrams 42—43, we may compare the conditions which are there found, with the above. The *real* range of expansion in the high pressure cylinder is 4·9, which will yield a smaller proportion of condensation,—other conditions being equal.

The proportion of *initial* condensation, relatively to the quantity of steam found at the termination, seems to be proportionate to the range of expansion in steam jacketed cylinders. Whether the proportion of *permanent* condensation maintains the same ratio, is altogether problematical. In view of all the known facts in connection with expansive working of steam, and in the absence of direct experimental data, it will be the best to assume provisionally, that the proportions of *permanent condensation* will be in the ratio of the range of expansion.—all other conditions being equal. Not only, however, is the range of expansion less in No. 42 than in No. 46, but the range of temperature to which the cylinder is subjected in the former, is less also,—being only 116° Fahrenheit, and equalling two-thirds of that of the latter. It is true that the full range of expansion in both cylinders of this Compound Engine is greater than in the case of Diagram No. 46. But, for the object immediately in view, we must confine our attention for a moment to the high pressure cylinder only, because the conditions which obtain here are those which determine the quantity of steam admitted above what can be detected by the Steam Engine Indicator, when judged in comparison with Diagram No. 46.

Assuming the condensation to be proportional to the range of expansion and temperature, and that the proportion in this case will be corresponding, or proportional to the Simple Engine just referred to, then as the ranges of expansion are 4·9 and 11·4, and the ranges of temperature as 2 : 3, so the ratios of condensation will be as 1 : 3·5. The permanent condensation in the case of Diagram 46 being 33 per cent., it follows that the permanent condensation in *Diagram 42* will be 9·4 per cent. of the steam admitted, and which

will be in the state of water held in suspension in the steam at the end of the stroke of the high pressure cylinder.

As 3·5 : 33 :: 1 : 9·4.

The terminal pressure here is 18lbs absolute, and in the low pressure cylinder it is 5lbs absolute; and the ratios of capacity being as 1 : 29, the quantity of steam at the latter pressure is only 88·73 per cent. of the former; of the steam remaining at the termination of the stroke of the first cylinder, we find then, that 11·27 per cent. disappears in the second. Of the whole quantity of steam which enters the first cylinder, 90·6 per cent. is found at the end; and of this, 11·27 per cent., equalling 10·21 per cent. of the whole, disappears in the low pressure cylinder. As 9·4 per cent. of steam is condensed in the high, and 10·21 per cent. in the low pressure cylinders, the total condensation will equal 19·61 per cent., so that the steam found at the end of the stroke of the second cylinder constitutes only 80·39 per cent. of the whole consumption.

When we compare this proportion of condensation with that of Diagram 46, and allow for an increase of expansion in the latter, so as to equal that of the Compound Engine (which is 14·21), the comparative condensation will be, for the Compound Engine 19·61 to 41 in the Simple Engine, or less than one-half. This conclusion is strengthened by the comparative amount of condensation in the Simple and Compound Engines represented by Diagrams 13 and 47—48, as found by computation on the best data available, and on those estimates which give the smallest difference between the two principles: see page 301. The correspondence which is thus found between the two comparisons of Simple and Compound Engines, in the proportionate difference of the total permanent condensation, is so close, that it is apparently something more than accidental. From this, and from other evidences which have been presented, we are fully warranted in concluding that—all other conditions being equal,—*the loss of steam by condensation will be twice as much in the Simple as in the Compound Engine.*

Let us now proceed to ascertain the weight of steam consumed per I.H.P. per hour of the Compound Engine under discussion. Calculated by the terminal pressure of steam (5lbs), the weight was found to be 10·915lbs per I.H.P. per hour. This having now bee

ascertained to constitute only 80·39 per cent. of the total consumption, then as

$$80{\cdot}39 : 10{\cdot}915\text{lbs} :: 100 : 13{\cdot}577\text{lbs},$$

which last may be regarded as the true weight of steam consumed per I. H. P. per hour in the Compound Engine, represented by Diagrams 42—43.

It will be interesting to carry the investigation a little further here, and determine, on the basis of the foregoing conclusion, the consumption of coal per I.H.P. per hour. The boiler—which is of the kind called "Lancashire," with two fire-box flues,—is set in the best-known way for securing economy. The flues, beyond the "bridge," are well filled with Mr. HEYWORTH's Copper Spiral Tubes, which give a great area of heating surface, and secure a rapid circulation of water and generation of steam. The boiler is so well covered, that the radiation of heat, either above or in front, is exceedingly small. The heat of the fires is so well utilized, that the temperature of the gases, after leaving the economiser, ranges below 300° down to 240° Fahrenheit. In view of all these conditions, the evaporative duty of the boiler should not be placed at less than 9lbs of water per pound of coal. As the consumption of steam is found to be 13·577lbs per I.H.P. per hour, then 13·577 ÷ 9 = 1·508lbs of coal per I.H.P. per hour.

In the descriptive analysis of these diagrams, the calculations made on other data, and by a different method, showed the consumption of coal to be 1·633lbs per I.H.P. per hour: see page 204. But on looking at the figures there given, it will be seen that the I.H.P. on which this result is based, is 160; whereas, the result just given above is on the higher power of 165 I.H.P. Substituting this latter amount, then 1·633 will be reduced to 1·583lbs of coal per I.H.P. per hour; so that the comparative results of the two processes of computation will now stand thus:

$$1{\cdot}508 : 1{\cdot}583 :: 95{\cdot}26 : 100.$$

The close agreement of the results of the two methods of computation, is presumptive evidence of their correctness, and increases the value of the conclusion arrived at. If the evaporative duty of the boiler be estimated at 8·5lbs of water per pound of coal, then the *results of the* two computations will be almost identical.

The important question will naturally arise, *What are the conditions and circumstances which should determine the adoption of the* COMPOUND ENGINE *in preference to the* SIMPLE ENGINE? In all cases where economy of fuel is of little or no consequence, and where, therefore, a high degree of expansion is not desirable, the Simple Engine will be more suitable, because less complicated and less costly. The compound principle will rarely be advantageous for Non-condensing Engines, because of the necessarily high terminal pressure, which should never be below the atmospheric line. In a Non-condensing Engine exhausting into the atmosphere direct, or through a feed water heater, it will be found more economical to lower the initial pressure, and extend the point of cut-off, when the load on the Engine happens to be reduced, rather than let the terminal pressure fall much below the atmospheric line. This statement will perhaps create some surprise. The simple explanation is this: When the terminal pressure is below the atmospheric line, the atmosphere rushes into the cylinder on the opening of the exhaust valve, and thereby causes a much larger initial condensation of the steam than would otherwise occur. This clearly limits the application of the compound principle to Non-condensing Engines (if applicable at all) to cases where a very high initial pressure constantly obtains. The lowest absolute initial pressure thus available would probably be, say 120lbs, with which the cut-off in each cylinder would require to be at 0·4 of the stroke, and the cylinders in the ratio of 1 : 2·5. This would give $2{\cdot}5 \times 2{\cdot}5 = 6{\cdot}25$ expansions. The initial pressure being 120lbs $\div$ 6·25 = 19·2lbs terminal pressure by the isothermal law. The variothermal law—making no allowance for fulcrum capacity—would reduce the terminal pressure to 16lbs, or 1lb above the atmospheric line. At 90lbs initial pressure, the steam might be cut off at one-fifth (neglecting fulcrum capacity), and the terminal pressure, by the variothermal law, would be 15·5lbs, or 0·5lbs above the atmospheric line. This would give a range of temperature in the cylinder of 105·5° Fahrenheit. Whether compounding under these conditions would be beneficial, is problematical.

THE ADVANTAGE AND ECONOMY OF THE COMPOUND ENGINE *is determined chiefly by the degree of expansion adopted.* What the lowest degree may be at which the principle will be advantageous, will be difficult to settle conclusively. It is not probable that any advantage would be derived from compounding with less than four

expansions. Beyond this degree, the inequality of pressure on the crank increases so much, that compounding is desirable to secure a useful equalization. Besides which, the range of temperature in the cylinder increases. The initial condensation also increases as already proved; and notwithstanding the greater consequent re-evaporation, the permanent, uncompensated condensation is absolutely greater as the cut-off is earlier, relatively to the measure of steam used. Furthermore, the proportion of steam permanently condensed in the cylinders is greater in the Simple than in the Compound Engine, for a given initial volume and pressure. The lowest degree of expansion at which compounding would result in increased economy, cannot at present be fixed. In the Table on Compounding at page 280, we have adopted 6·666 expansions as the ratio at which the principle may in all cases be safely applied. From this degree down to four expansions, the gain by compounding will probably diminish rapidly.

The calculations with regard to the amount of power, in the Table on Compounding, having been made on the basis of a piston speed of 500 feet per minute, it may be presumed that we advocate such speed as the most suitable and proper. It will perhaps be advisable, therefore, to examine *the question of Speed in its relation to economy, steadiness, and general utility.*

Many and varied circumstances and conditions influence the determination of the speed. The highest practicable speed which can be permanently worked without serious deterioration of the Engine, and without loss by the generation of back pressure in excess of what would occur at a lower speed, will generally be desirable. The advantages of high speed will be,—first, that less gearing will be required to obtain the ultimate speed of the machinery; second, that a greater amount of power will be obtained from a given size of Engine; third, that greater uniformity of rotation will be secured; fourth, that less condensation will take place in the cylinders, because less time is given for the cooling of the metal during exhaustion, and whilst the steam is at low pressure.

Let us suppose that we have one Engine with a cylinder of 5 feet stroke, with an angular velocity of 50 revolutions per minute, and driving, say 200 I.H.P.; and let us suppose that we have another Engine exactly similar, but running at a speed of 25 revolutions per minute, and having the same initial pressure and measure of steam. *Now it is quite* clear, in the first place, that the latter Engine would

not produce more than half the power of the first; secondly, that its external radiation would be greater relatively to the amount of power; and thirdly, that it would produce a larger amount of condensation at each stroke of the piston. The first and second results here stated are self-evident, and anything more said concerning them would be superfluous. The third result stated may not be quite so clearly evident to every reader.

On the admission of the steam to the cylinder at a given pressure and temperature, the metal will be raised in temperature by contact and convection; and the abstraction of the heat of the steam by the metal will continue until, by conduction, the outer surface of the cylinder shall have acquired the same temperature, or nearly so, as the inner one—if time sufficient shall elapse; beyond this, the heat transmitted to the metal by the steam will simply be the equivalent of external radiation. During the time of exhaustion, the internal surface of the cylinder, being exposed to the action of steam of lower temperature and high absorbing power, is reduced in temperature thereby. With these conditions, the abstraction of heat from the cylinder will inevitably be greater when the time is greater; but whether it will be exactly in the ratio of the time, is not certain. We need not doubt, then, that the condensation at the higher speed will be less for each stroke of the piston, and therefore that the higher speed will be the more economical.

HIGH SPEED tends to produce a loss which it is important not to overlook,—that of excessive back pressure. If, by a higher speed, there should be a saving of, say 5 per cent., by the smaller proportion of condensation, but a greater uncompensated back pressure, equivalent to 10 per cent., then clearly there would be a balance on the side of loss. The back pressure resulting from any given speed of piston will be independent of the length of stroke—all other conditions being equal. For example: If one Engine should have a stroke of 5 feet, and run at 50 revolutions per minute; and if another Engine should have a stroke of 2 feet 6 inches, and run at a speed of 100 revolutions per minute; and if both cylinders should be of the same diameter, and have exactly the same area of valve orifice during every part of the stroke—or, what would amount to the same thing, have the same proportion of area of orifice to the sectional area of the cylinder; and if, furthermore, the steam should have the same terminal pressure in each case; then, undoubtedly, the back pressure would be the same

in each case—so far, at least, as the conditions of the Engines here indicated govern the results. As in the first case there will be double the time for the steam to escape, yet, as there will be double the quantity for each stroke, so there will necessarily be the same amount for each unit of time; and therefore the two cases will be on an equality in this respect. *The result is governed by the speed of piston, the proportion of the valve orifices to the area of the piston, and the pressure of steam to be transferred from the cylinder to another space.*

In the case of a *Compound* Engine, where the steam, in being transferred from the high pressure cylinder, passes at once to the low pressure cylinder through two sets of valves, and through contracted pipes—perhaps having several angles,—the loss of pressure due to friction will be considerable, if the speed be high. By having a RECEIVER between the high and low pressure cylinders, *of large capacity*, the transmission of the steam from the first cylinder to the second, will be effected with less loss (that is, with less difference of pressure), because there will be less obstacles interposed to the free passage of the steam.

When the ports and valve orifices are made of the largest practicable area, a piston speed of 500 feet per minute will be found not to generate unnecessary back pressure. By *necessary back pressure*, must be understood the pressure of the *medium* into which the steam shall exhaust, and below which it cannot possibly fall. Such, for example, as the pressure of the atmosphere, into which a Non-condensing Engine shall exhaust. When the high pressure cylinder of a Compound Engine exhausts into a *Receiver* (this latter having an approximately constant equilibrium of pressure), the back pressure should coincide with it, or as near as possible. Between the pressure in the *Receiver* and the initial pressure in the second cylinder, there will necessarily be a *slight* difference, resulting from the condensation of the steam on its entrance into the cylinder. This part of the subject has been treated more minutely in an earlier part of this chapter: see pages 287—288. If the difference be anything considerable, then it may be safely concluded that the valve orifices are insufficient to permit the free transmission of the steam; or, what will be equivalent, that the speed is too great to permit the steam to pass with sufficient rapidity, with such orifices as exist.

It must not be concluded that a piston speed of 500 feet per minute is advisable for any and every length of stroke. A high

speed of the fly wheel undoubtedly is an advantage,—in two relations. In the first place, as already observed, a greater speed in the prime mover will give the ultimate speed required with less and lighter intermediate gearing and shafting, for the transmission of the motion; and in the second place, the fly wheel will be increased in its *inertia* in the ratio of the squares of the speeds. On the other hand —as we have seen in treating on the inertia of the piston, &c.— when the speed is high, the inertia of these parts becomes so great that the difficulty of equalizing the forces is greatly increased, and the greater wear and tear (as ordinarily expressed) of the Engine is the result. There has almost invariably been great difficulty experienced in the working of Engines at high piston speed, unless the stroke has been of good length, so as to avoid excessively rapid reversals of the piston, which so frequently produce heating of the crank pin and journal. This effect is not produced to the same degree in the Compound as in the Simple Engine—all other conditions being equal—because of the greater equality in the incidence of pressure. This difference has been experienced notably in the case of the Simple and Compound Engines represented by Diagrams 13, and 47—48.

The extreme importance and interest of the subject of Compounding, and the numerous features which were necessary to be introduced and elucidated, in order to make the treatment as complete and satisfactory as the available data permitted, have caused this chapter to be extended to such considerable length, that it will be advisable, in concluding, to give a recapitulation of the essential points discussed and determined,—with what success the reader will judge. The conclusions which have been arrived at may now be presented in the following congeries of statements:

1st.—The Compound is to be preferred to the Simple Engine for a high degree of expansion, because the pressure on the crank can be better distributed.

2nd.—A large proportion of Compound Engines hitherto made have the cylinders disproportioned; and, as a consequence, do not yield the full duty of the true principle of expansion.

3rd.—The law of correct proportions of the cylinders of Compound Engines has not heretofore been determined, as is evidenced by the great diversity in practice.

4TH.—The Rule of Proportions for Compound Engines, given on page 231, will give the most perfect expansion of the steam, and the highest co-efficient of expansion of the diagrams.

5TH.—Practically, with proportions of cylinders which do not give the full theoretical duty, the Compound is more economical than the Simple Engine.

6TH.—The Compound Engine, proportioned and arranged according to our "Rule," will give the highest attainable duty, theoretically and practically.

7TH.—The Compound Engine, on our plan, will give, in actual working, an aggregate effective pressure on the pistons at half stroke, of *double* that which is given by the Simple Engine, for the same initial measure and pressure of steam, and therefore for the same range of expansion.

8TH.—The pressure on the crank is directly as the pressure of steam on the piston or pistons,—whatever may be the weight or speed of the reciprocating parts,—when near the middle of the stroke, and when the crank and connecting rod form an angle of 90°.

9TH.—The distribution of the aggregate energy of the steam exerted on the pistons, being in the Compound Engine in so much higher proportion at the angle of 90° of the crank, than in the Simple Engine, is more usefully exerted.

10TH.—No possible arrangement of weight and speed, can so adjust the inertia of the reciprocating parts in the Simple Engine, as will give the same useful incidence of pressure on the crank pin which can be obtained in the Compound Engine,—unless the range of expansion be very small.

11TH.—The Direct Acting Vertical Engine, as hitherto made, is objectionable, inasmuch as it is not correctly balanced, and as a consequence, generates unsteadiness in working.

12TH.—The Compound Engine is not subject to heating of the crank-pin and journal to near the same degree as the Simple Engine,—both having the same range of expansion, and all other conditions being equal.

13TH.—The initial condensation in any cylinder is greater relatively to the measure of steam admitted, as the cut-off is earlier.

14TH.—The permanent condensation in the Simple Engine is

double that of the Compound Engine,—all other conditions being equal.

15TH.—With the same proportion of permanent condensation to power obtained, the gain by increased pressure of steam will be as shown in lines 17 and 21 in the Table of Proportions, &c., for Compound Engines, page 280; assuming—what is almost invariably allowed—that a given weight of water will be evaporated into steam of any given practicable pressure by the same weight of coal. *

16TH.—By squaring the number of expansions, the power will be doubled with the same weight of steam,—provided that the permanent condensation shall bear the same proportion to the power obtained, and that the terminal pressure in the low pressure cylinder be coincident with the back pressure there.

17TH.—By the application of a RECEIVER between the high and low pressure cylinders, the true principle of expansion can be carried out in the Compound Engine.

18TH.—By the use of a Receiver, the "GAP" between the high and low pressure cylinders, so often regarded as inevitable in the Compound Engine, may be obviated; and the coincidence of the back pressure in the first, and the initial pressure, or steam line, in the second cylinder, almost completely attained.

19TH.—By having the Receiver of large capacity, the pressure therein will approximate to a constant equilibrium; and the exhaust line in the high pressure cylinder, and the initial pressure line in the low pressure cylinder, will, or may, be parallel with the atmospheric line.

20TH.—In any Compound Engine, where the two cylinders are placed on parallel lines, the use of a Receiver will permit the cranks to be fixed at right angles; and such positions will secure a better distribution of the rotating efforts, and consequent greater uniformity in the motion of the fly wheel.

21ST.—By our rule of proportion for Compound Engines, each cylinder will yield its proper and most useful share of the total power,—the low pressure cylinder always giving the greater power; but the difference between the two cylinders

* By increasing the boiler pressure from 50lbs to 100lbs per square inch by the pressure gauge, one per cent. more heat will be absorbed in evaporating a given weight of water from an initial temperature of 50° Fahrenheit.

never being greater than is shown in the Table of Proportions, &c., and any deviation therefrom will be in the closer approximation to equality of power; yet still leaving to the low pressure cylinder more than half the total power, if the valves continue to cut off at the points fixed by rule.

22ND.—The Compound Engine is more economical than the Simple Engine, inasmuch as a less consumption of fuel is required, and a less increment of temperature given to the condensing water, for the same given amount of power,—all other conditions being equal.

23RD.—The speed of piston should be as high as is practicable, without generating unnecessary back pressure, or great inequality in the incidence of pressure, or unsteadiness in working, or excessive wear and tear.

24TH.—Stated concisely and comprehensively, the Compound Engine, when proportioned and arranged according to our rule, &c., will give a better distribution of pressure on the crank, or cranks; will not produce such dangerous strains; will give greater steadiness and uniformity of rotation, more especially if there be two cranks placed at right angles; will give a *double effective pressure* on the pistons at half-stroke, when the cranks are at right angles with the connecting rods, and inertia inoperative; will generate less heat in the bearings, and will therefore dissipate less energy externally; will produce less permanent condensation of steam in the cylinders; will transmit less heat to the condenser, and therefore will consume less heat, less steam, and less fuel,—as compared with the Simple Engine giving the same amount of power, and having the same initial pressure and range of expansion,—and all other conditions being equal.

W. S.

CHAPTER XI.

STEAM JACKETING.

THE exact value or benefit of Steam Jacketing has not yet been determined in such a manner as to set the question at rest, and furnish a sure and unerring guide to the Engineering world. The subject seems to be shrouded in uncertainty and mystery to such a degree, that many deny altogether that any saving can be effected by the use of the Jacket. Not long ago, at one of the meetings of the Institution of Mechanical Engineers, this question was debated by the members present, after the reading of a paper treating upon it, and some of them declared that they had proved experimentally that no economy whatever resulted from the application of steam to the Jacket of the cylinder. Such being the position of the question, it is important to examine it by the light of accurate and well authenticated data. Whether or not we shall be able to arrive at clear and definite knowledge of the laws which govern the operations, we may nevertheless ascertain the general conditions which will secure good and important results.

That there is economy in Steam Jacketing when properly applied, can be proved satisfactorily. The simplest and clearest proof, and one which can be understood by the commonest apprehension, is supplied by the case of a Steam Engine, the cylinder of which is Jacketed, and the Jacket supplied with steam direct from the boiler. With steam in the Jacket, the Engine could be run at the proper speed without difficulty; but when the steam was shut off from the Jacket, the speed could not be maintained, and a larger consumption of fuel was required. When the steam was again allowed to flow into the Jacket or Casing, the proper speed was soon regained, and less stoking was required. Numerous examples, corresponding with this case, support the same conclusion.

A case of Steam Jacketing at Mr. ELI HEYWORTH'S, Audley Hall Mill, Blackburn, Lancashire, is very noteworthy. The experiments with this Engine, which have been numerous and exact, furnish some of the most important and valuable data. The Engine is a Compound

Horizontal, with the high and low pressure cylinders placed parallel with each other,—the pistons being connected with separate cranks, which are placed at opposite ends of the fly-wheel shaft. The cylinders are both Cased or Steam Jacketed, and these Jackets receive steam direct from the boiler, the pressure in which is 80lbs per inch by the pressure gauge. The condensed water is conducted back to the boiler direct, so that there may be the least possible loss of heat. A more detailed description of this Engine is given in connection with Diagrams 42 and 43.

Three distinctly different kinds of tests have been made with these Steam Jackets, in order to ascertain their efficiency :

1st.—The difference in the speed of the Engine, *with* and *without* steam in the Jackets ; every other condition remaining unchanged.

2nd.—The quantity of steam admitted into the cylinder at each stroke, *with* and *without* steam in the Jackets; whilst the weight or load, speed, boiler pressure, and every other condition remained unaltered.

3rd.—The increase of temperature given to the injection water by the steam which passes through the cylinders into the condenser, *with* and *without* steam in the Jackets ; whilst the speed, load, boiler pressure, quantity of injection water, and all other conditions remained constant.

With regard to speed, the first of the series of tests enumerated above, a detailed account will be found in description of Diagrams 42 and 43. It will be sufficient to state here, that the increase of speed was from 33 revolutions per minute *without* steam in the Jackets, to 37½ revolutions *with* steam in the Jackets. Here is a remarkable difference of speed due to the influence of steam in the Jackets. Whether this increased speed is obtained by the same consumption of steam as the slower speed, is problematical, and deserving of close investigation. The question will be found treated in connection with Diagrams 42 and 43.

The next test relates to the amount of steam used at each stroke of the piston, *with* and *without* steam in the Jackets, as shown by the Indicator Diagrams ; and as detailed particulars are given in the treatment of these, it will only be necessary to say here that, with steam in the Jackets, 10 per cent. less is admitted into the cylinders, as shown by the pressure at the termination of the stroke in the high

pressure cylinder (which takes no account of the quantity then held in the state of water), than when the Jackets are not thus supplied, and when the loss by condensation in the Casings has been allowed for,—the loss here being small.

The third test made was to ascertain the amount of heat which passes from the cylinder to the condenser. This admits of exact demonstration, and supplies the most reliable and valuable data by which to determine the efficiency of Steam Jacketing. A considerable number of tests have been made, and with a uniformly good result.

With the speed, the boiler pressure, the weight on the Engine, and the quantity of injection water remaining constant, the increase of temperature of the injection water, when steam is in the Jackets, is 26°; whereas, when steam is not in the Jackets, the increase of temperature is 30° Fahrenheit. Say, for example, that the injection water in both cases is 50° initial temperature, then the ejection water will be in the one case 76°, and in the other case 80°. The experiments have been made on different days, and at various hours of the day; and the initial temperature of the injection water has een somewhat variable, as must obviously be the case; yet the increase of temperature given to the water by the steam, has been in every instance as above, within the smallest limits of variation. The slight difference which has occurred would make the mean of the differences between the two conditions a little more than is given above, but so little in excess that the variation may be entirely disregarded. We may, without hesitation, state that the amount of heat passed into the condenser when steam of 80lbs pressure is supplied to the Jackets of the cylinders, is only 86·6 per cent. of the amount thus passing when the Jackets are not supplied with steam. Here there is a saving of 13·4 per cent. of heat, at the cost of the condensation of steam in the Jacket.

The experiments, with a view to ascertain the amount of condensation in the Jackets, seemed to prove that the loss of heat here is exceedingly small: see detailed account of this in description of Diagrams 42—43.

An important element in the case has yet to be introduced, v the degree of expansion. Here the real expansion amounts to 14· and the apparent expansion—that is, as shown by the point of c off in the diagram—is 18·85 times. With a less degree of expans

the saving by the application of Steam Jackets would probably be proportionately less.

We have here given the facts of a series of different tests, and they give so close an approximation to uniformity of result, that little doubt can remain of the general correctness of the conclusion to which they unmistakably point.

The results of Steam Jacketing of the cylinders of the Beam Engines of SIR TITUS SALT, BART., SONS, & CO., Saltaire, have been detailed in connection with Diagrams 45—46. The real ratio of expansion in Diagram 46, which represents the condition of working of the Engines during the time of making the extensive series of experiments, and which therefore serves as the basis of the calculations in connection therewith, is 11·4, whilst in the case just examined the ratio of expansion is 14·21. The cylinders, in the present case, be it remembered, are surrounded with steam at both ends, as well as at the sides, which will make a considerable difference in the area of surface (internal) thus influenced by external heat. If it be beneficial to heat the cylindrical part of the cylinder, then obviously, it will be still more beneficial to heat every part. And if the piston itself could be supplied with steam internally, then the beneficial results must be still further increased.

The opinion has been held by some very eminent authorities, that the beneficial results of Steam Jacketing were much greater in the Simple than in the Compound Engine; that indeed the gain, if any, in the Compound Engine, was exceedingly small. Now the Compound Engine represented by Diagrams 42 and 43, and the Simple Engines represented by Diagram 46, furnish an excellent means of comparison. There is no great disparity in the ratio of expansion, and the Jackets in both cases are supplied with steam of the boiler pressure. Although the Jackets of the Compound Engine are supplied with steam of a higher pressure, and therefore of a higher temperature, than the Simple Engines, yet the former has the greater ratio of expansion, and the cylinders have not the whole internal surface heated by the Jackets. The temperature of 50lbs absolute, as in the case of the Simple Engines represented by Diagram 46, is 281° Fahrenheit; and in the Compound Engine, with 95lbs absolute, the temperature is 324°. Taking all the conditions of the two cases into account, the advantage will be on the side of the Simple Engines in the proportionate benefit to be derived from Steam Jacketing.

What are the actual results? In the Simple Engines the saving amounts to 14·3 per cent., and in the Compound Engine to 13·3 per cent., as near as the data enable us to determine.

This comparison may not be absolutely correct, for the following reasons: In the case of the Compound Engine (Diagrams 42—43) the gain of 13·3 per cent. is deduced from the tests having reference only to the quantity of steam passing through the cylinders, and not including that supplied to the Jackets. In the case of the Simple Engine (Diagram 46) the saving of 14·3 per cent. is on the whole consumption of steam; but when calculated for *that* steam only which passes through the cylinder, then it will amount to 17·16 per cent. Is the quantity consumed in the Jackets in the same proportion to the gross consumption in the two cases, or even approximately so? Our trial to ascertain the quantity of steam condensed in the Jackets of the Compound Engine showed it to be exceedingly small. Still, the result of this one trial cannot be accepted as settling the question of the proportionate amount of condensation in the Jackets of Compound Engines. Careful experiments are required to determine the proportion of condensation which may occur in the Jackets of Simple and Compound Engines,—all other things being equal.

That a larger proportionate consumption of steam is required in the Jackets in the case of the Simple Engine (Diagram 46) than in the case of the Compound Engine (Diagrams 42—43) here under consideration, is rendered highly probable by the tendency of many facts. In the first place, as the ends as well as sides are Jacketed in the Simple Engine, there will necessarily be a larger abstraction of heat from the steam in the Jackets, than when the cylindrical part only is cased. And in the next place there will probably be a larger transmission of heat from the covers to the steam within the cylinder for each unit of area, than from the cylindrical and bright parts of the cylinder. More of this hereafter. In the third place, the Simple Engine will abstract a larger quantity of heat from the steam in the Jackets than the Compound Engine,—other conditions being equal. This last statement is fully warranted by the fact that the Compound Engine carries less heat into the condenser than does the Simple Engine, for the same amount of power and the same range of expansion, and all other conditions being equal. For evidence of this, see page 240. As a larger quantity of heat passes through the cylinders of Simple than of Compound Engines for a given amou

of power; and as the initial condensation and subsequent re-evaporation is greater in the former than in the latter, whether the cylinders are Steam Jacketed or not; then it follows that a greater alternation of temperature of the metal takes place in the case of the Simple than of the Compound Engine, and that therefore a larger abstraction of the heat of the metal by the steam is the result. Granting this, then evidently the steam in the Jackets will have a correspondingly greater quantity of heat to communicate to the inner surfaces of the cylinders with the simple than with the compound principle. In view of all the conditions and demonstrated tendencies enumerated in this connection, it may be safely concluded that there will be a much larger condensation in the Jackets of the Simple Engines (Diagram 46), than in the Compound Engine (Diagrams 42—43), in proportion to the amount of power. How much more cannot be determined now.

The Simple Engines having the smaller number of revolutions per minute, even though of greater piston speed, will have the internal surfaces exposed to the cooling influence of the condenser for a longer time, and will therefore be subjected to a greater abstraction of heat per unit of surface,—other conditions being equal. The range of temperature is slightly greater in the Compound than in the Simple Engines, taking, in both cases, the initial and the final back pressure temperatures. As in the case of the Compound Engine there is a fall of 10lbs from the boiler pressure to the initial pressure in the cylinder, a slight advantage will be obtained independently of the Steam Jacket, as will be explained further on, and somewhat lessening the proportionate gain from the Jacketing.

A striking illustration of the effect of the steam in the Jackets of the cylinders of SIR TITUS SALT, BART., SONS, & Co., Saltaire, is the fact, that if the steam be shut off from the Jackets for a short time, and unknown to the stokers, they discover, by the heavier firing required, that something is amiss, and soon begin to inquire what is the matter.

The difference in the increment of temperature given to the injection water corresponds approximately with the proportion of saving in fuel.

As in the chapter on Compounding we have shown, by a notable and reliable case (see page 239), that by compounding, a saving of 33 per cent. was effected then presumably, an equal saving would be *effected in this* case. But let us assume the gain by compounding at

25 per cent. on the higher consumption of fuel. This would reduce the 84 tons per week, to 63 tons; and allowing a still further reduction of 14 per cent. by Steam Jacketing, the consumption would then be 54·18 tons per week. As the saving effected by compounding, from Diagrams 13 to 47—48, amounts to 33 per cent. with 6·2 expansions, then it may safely be concluded that, with 11·4 expansions, as found in Diagram 46, the saving would not be less, but more; because, by the facts and arguments advanced in the chapter on Compounding and elsewhere, it is clear that the proportionate gain in changing from the simple to the compound principle, will be greater with the higher ratio of expansion. What may be the ratio of increase in economy relatively to the higher ratio of expansion in changing from the simple to the compound principle—all other conditions remaining the same,—we have not the data which will enable us to determine.

The reader will remember, that of the estimated saving of 33 per cent. just referred to, a part of this, amounting to 9 per cent., was stated to be due to the higher co-efficient of expansion which is found in the Compound Engine, leaving 24 per cent. as the net gain when corrected to the same co-efficient of expansion, and for the same amount of power, by the method described on pages 304—306: see also page 242.

The Simple Engine (Diagram 46) with 11·4 expansions, would, in like manner, if compounded, give a higher co-efficient of expansion, with the initial pressure and point of cut-off remaining as at present, because the terminal pressure, instead of being 6·75lbs as in the Simple Engine, would then be only about 4lbs. This would seem to give a greater range of expansion; and certainly, when estimated by the initial and terminal pressures, such would be the case. When, however, the initial and final volumes only are considered, the range of expansion will remain unchanged. The difference in the co-efficients being the result of the greater initial condensation and subsequent re-evaporation which occurs in the cylinder of the Simple Engine.

The increase in the co-efficient of expansion would not, however, be so much as in the case of changing from Diagrams 13 to 47—48, because there the increased cylinder capacity gives a greater range of expansion than before, and therefore the gain to be expected in the co-efficient of expansion by compounding No. 46, would not be so great. It does seem safe then to conclude that an equal gain would

result by compounding the Simple Engines represented by Diagram 46, as was experienced by changing from Diagram 13 to Diagrams 47—48, and that therefore when we assume 25 per cent. as the saving to be effected by the compound principle, we are fully warranted by an example which furnishes the most abundant proof.

As the calculations in the chapter on Compounding in connection with Diagrams 13 and 47—48, show the permanent condensation in the former to be 43 per cent., and in the latter only 15 per cent.; or, by the higher estimate of evaporative efficiency of the boilers, to 49·25 per cent. and 25·25 per cent. respectively (see pages 242 and 301); and as the initial condensation has been proved to be greater as the ratio of expansion is greater,—all other conditions being the same,—then, it is probable that the permanent condensation is greater also.

Adopting this last conclusion hypothetically, it will explain the cause of the disappointment which is so generally experienced in working with a high degree of expansion in the Simple Engine. We find many cases of very moderate expansion, — say with initial pressure of 30—35lbs absolute, and a terminal pressure of 5—7lbs, and having no very decisive cut-off, working almost, if not quite, as economically as some of what are looked upon as the best examples of high initial pressure, and early and instantaneous cut-off. We will assume the boiler pressure to be the same in these cases just introduced, notwithstanding the great difference in the initial pressure on the piston, and range of expansion. This pressure we will say is 60lbs by the gauge, making 75lbs absolute. We will assume the early cut-off to have an initial pressure of 70lbs, and the later cut-off to have an initial pressure of 35lbs absolute. The temperature due to 75lbs boiler pressure is 307·5°. Neither of the cylinders are supposed to be Steam Jacketed.

When steam falls in pressure without resistance being opposed to it, and without performing work, no loss of heat takes place, except so much as may result from contact with the sides of the vessel into which it may issue; so that if such vessel should have the temperature of the steam entering, there will be no sensible change; hence, when steam of 75lbs pressure in the boiler, passes into the cylinder at 35lbs, it has a temperature of 48° in excess of that due to the latter pressure. This excess of temperature is expended in supplying heat *to the cylinder* which has been reduced in temperature by being

exposed to the cooling influence of the condenser. Until the whole of such excess of heat has been given up by transmission to the metal, and by transmutation into the motion of the piston, no condensation can take place, the result of which is, that the initial, and probably the permanent condensation is proportionately less.

So far this seems clear, and the conclusion is supported by the highest authorities. As, however, the temperature of the metal of the cylinder will certainly be reduced as low—all other things being equal—in the one of low initial pressure, as in the one of high initial pressure, it is not apparent at once why the steam—which is assumed to be of the same temperature in both cases, although of different pressures—should be more condensed in the one cylinder than in the other. The difference in the pressures constitutes an element in the case, the influence of which seems to have been overlooked by all the writers on this question. The elements which must be recognised in the investigation are,—the temperature of the steam, its pressure, volume, and the temperature and area of the internal surface of the cylinder to the point of cut off—including cover and face of piston.

If the steam in its expansion in the cylinder of the Simple Engine be assumed to correspond to the hyperbolic curve, then, for the same terminal pressure, the higher initial pressure will give the greater power, and its point of cut-off will be earlier in proportion as its pressure is higher. If, however, an excess of initial and permanent condensation takes place in the case of higher pressure and earlier cut-off, greater than the gain which results from the higher ratio of expansion as shown by the hyperbolic curve, or isothermal law, then, clearly, the balance will be in favour of the lower initial pressure. This is just one of the questions which requires to be determined. The law of expansion, as showing the increase in the power by an increase in the ratio of expansion, can only be a correct representation of the relative economy, when the condensation bears the same proportion to the weight of steam used per stroke of the piston. As the point of cut-off is earlier, the internal surface of the cylinder with which the steam is in contact to such point of cut-off, increases in an accelerating ratio relatively to the volume of steam at initial pressure. Hence the earlier the cut-off in a cylinder, and the greater is the proportional area of the surface with which the steam has come in contact—and as a consequence has been condensed,—relativel to the whole internal surface of the cylinder,

If, in one case, we have a cylinder in which the absolute initial pressure is 35lbs, and the terminal pressure 7lbs, then the cut-off will be at one-fifth of the stroke; and if the stroke should be 5 feet, the cut-off will occur at 12 inches traverse of the piston. If, next, the initial pressure in the cylinder be 70lbs, which is double that of 35lbs, then the cut-off will require to be at one-tenth of the stroke for the same terminal pressure. Say the diameter of the cylinder is 30 inches, which will give an area of 706 inches, and a circumference of 94·25 inches. As the diametrical or sectional area of the cylinder includes both cover and piston, then the surfaces will be $706 \times 2 = 1412$ inches; and adding the cylindrical or circumferential area, which is for the lower initial pressure and later cut-off, $94{\cdot}25 \times 12 = 1131$ inches, will together give 2,543 inches. When the cut-off is at one-tenth (6 inches), the total internal area at the point of cut-off will be 1,977·5 inches, being 77·76 per cent. of the former, or seven-ninths. The real difference will be still less than this, as the surface area of the ports, &c., will be the same whether the cut-off may be earlier or later.

Let us come now to the consideration of the effect of pressure. We will assume that the cylinder has been reduced to the same temperature during the exhaustion of the steam, and that the temperature of the steam on its admission to the cylinder is the same in both cases, as shown above, even though the pressure in one case is double that of the other. Now the rapidity with which the heat of the steam will be transmitted to the metal will be as the pressure, —all other conditions being equal. The heat of the steam being transmitted to the metal by convection; and the quantity thus transmitted being as the velocity of the fluid passing over the surface in a given time; and the measure of the velocity of the atoms of the steam being as the pressure; it follows that the transmission of heat will be as the pressure. Then the conclusion seems clear, that the condensation at the higher pressure will be greatest for a given time and area. As, however, the lower pressure must heat a larger surface in the ratio of 9 to 7, then the amount of heat transmitted will be supposed to be increased from one-half in the ratio of 7 to 9, and that therefore the initial transmission of heat will be increased from 50 to 64·3 per cent.

There is one more element which has not yet been introduced into *the consideration* of the question. This element is *time*. It is quite

clear that time will be an important element in the case, until such duration as will have sufficed to produce an equilibrium of temperature between the steam and the metal. The lower pressure will have the longer time for the transmission of heat to the point of cut-off, but not double,—the time being as 1 : 1·4324. This would seem at first sight to bring the value of the two cases to near an equality, because 64·3 per cent. multiplied by 1·43 will equal .92 per cent. *This view would be altogether fallacious.* The time, to any given point of traverse, is the same in both cases, although in one case the heat of the steam is transmitted to the metal before the cut-off occurs, and in the other case after, to a given point of traverse ; so that although the element of time, as here stated, may have some influence, it is probably only small.

The permanent effect of the initial condensation will be considerable, even though there be a plentiful re-evaporation as the steam expands, because of the much greater conductivity of watery than of dry steam. When the steam has once become charged with water, as the result of condensation or otherwise, its capacity for carrying away the heat is immeasurably increased. This is abundantly and conclusively proved by the fact that when the steam, before entering the cylinder, is superheated, its terminal pressure and temperature are lower, and much less heat is transmitted to the condenser, as shown by the less increment of temperature given to the injection water. If, when steam of a higher temperature passes into the cylinder, a lower temperature, and also a smaller quantity of heat, issue from the cylinder to the condenser, then it is clear that the *state* or *condition* of the steam, due to superheating, is not so efficient in conducting and distributing the heat of the metal, and hence less heat is transmitted to the condenser.

The smaller sum of heat carried into the condenser is correlated with the fact that, when the steam is superheated, a smaller quantity suffices to produce a given amount of power ; or rather, the smaller quantity of heat carried into the condenser, and the smaller quantity of steam required for a given amount of power, are two different phases of the same operation, and which resolve themselves into reciprocal concomitants. Under the conditions here defined, the fall in temperature, which must inevitably occur by the performance of work, does not result in the liquifaction of the steam, except to very small extent ; it cannot therefore take up so much of t

heat of the metal before passing into the condenser, as when it is heavily charged with vapour, or mist.

If the reasoning and conclusions arrived at so far be correct, then the alternations of temperature of the metal of the cylinder will not be so great : and indeed, this conclusion is a corollary of the former one; because, if with a higher initial and a lower terminal temperature, less heat passes out of the cylinder, then the alternations of temperature occur more in the steam, and less in the metal. *This is the result which should always be aimed at,—to have the smallest possible alternation of temperature in the metal, and the greatest in the steam.*

Seeing that it is of such extreme importance to prevent the liquifaction of the steam, which may place in contact bodies of such high conductivity as water and iron, and which would as a consequence carry off the heat of the metal of the cylinder into the condenser; and remembering that dry steam has only small conductivity, in comparison with watery steam, it will be apparent that if the inner surfaces of the cylinder can be maintained at such a temperature as will prevent condensation on the admission of the steam, and also re-evaporate that which may be condensed by the performance of work, it will have a beneficial effect. If possible, there should be no more condensation on the surface of the metal than is necessary to assist in the lubrication. The beneficial results here under consideration may be produced, to some extent, by the application of the Steam Jacket. The extent to which such effect may be produced, will depend on the pressure and temperature of the steam admitted to the Jackets,—the proportion of the whole internal surface of the cylinder thus surrounded,—and likewise on the thickness of the metal through which the heat must pass from the Casing to the internal surface. Iron being in all cases the metal used, the conductivity will be the same,—other conditions being the same.

To include, as just above, pressure, apart from temperature, as an element which contributes to the effect of the steam in the Jacket, may perhaps seem superfluous; but remembering the remarks on pressure and temperature in a slightly different connection, it will be seen that the transmission of the heat of the steam in the Jacket to the metal, will be, not alone as the temperature, but as the pressure also; because, if the steam should be superheated, as urged in another part of this chapter, then its temperature will be higher than that

due to saturated steam of the same pressure. The increase of pressure by superheating will be very small,—being represented by the co-efficient of expansion for dry elastic gases. Superheating, properly speaking, cannot come into operation until every particle of water which may be held in suspension, has been turned into dry, anhydrous steam. When this point has been reached, the law of expansion of gases by heat comes into operation, which gives equal increments of volume by equal increments of temperature at a constant pressure. The co-efficient of increase of volume is 0·002035 for one degree Fahrenheit.* Taking the steam of 75lbs absolute pressure, the temperature of which for saturated steam is 307·5°, and superheating it to the temperature of 350°, its pressure would be increased to 81·5lbs, whereas saturated steam of the same temperature would have a pressure of 135lbs.

The thickness of the metal of the cylinder, that is, between the steam in the Jacket and the internal surface of the cylinder, is of very great importance, because the transmission of heat, both in time and quantity, is inversely as the distance passed through; and therefore the shorter such distance, the more efficacious will be the steam in the Jackets. It would be well, with this view, to cast the working cylinder as thin as practicable, and give strength by external ribs, at suitable distances apart, longitudinal and transverse. An objection to this plan will be in the impossibility of repeated re-boring of the cylinder. But this small advantage had better be sacrificed to permanent economy of steam. Besides, it would be practicable so to harden the internal face of the cylinder that little wearing would occur. And the harder the surface, the less would be the friction. This being done, the advantage would be considerable.

It has been shown in connection with Diagrams 45—46, that notwithstanding the application of steam of the boiler pressure to the Jackets, the initial condensation is very considerable. This, in all-probability, is caused in a great measure by the great range of temperature to which the internal surface of the cylinder is exposed, —the initial pressure of steam, which is the boiler pressure, or very near it, representing 281° Fahrenheit, and the terminal part of the exhaust line representing about 92°, giving a difference of 189°

* It has been established experimentally that gases increase one two-hundred-and-seventy-third of their volume at a constant pressure, for one degree Centigrade increase of temperature, and this is equivalent to one four-hundred-and-ninety-one decimal four for one degree Fahrenheit, and will therefore be correctly expressed decimally by 0·002035.

between the steam admitted and that which has just departed.* If the steam in the Jackets was sufficient to maintain the internal surface of the cylinder at the temperature corresponding to the steam at initial pressure, then clearly, no condensation would occur, except as the result of work performed, and this would be immediately re-evaporated; or, at least, such portions as should come in contact with the heated metal.

As the temperature of the steam supplied to the Jackets is the same as that supplied to the cylinders (in the case represented by Diagram 46), then it is simply impossible that the internal surface can be of the same temperature, because it has been exposed to the cooling influence of the condenser during the whole of the return stroke, and the inevitable result will be a lower temperature than exists within the Jackets. With steam of higher pressure and temperature supplied to the Jacket, and the metal between this and the inner surface thinner as suggested just above, the initial, and also the permanent, condensation would probably be considerably reduced. The transmission of heat by conduction is governed by time and distance; and the amount transmitted will be greater as the distance is less. It is clear then that to secure the greatest attainable economy by Steam Jacketing, these must be supplied with steam of a much higher temperature, and higher in proportion as the metal is thicker through which the heat must be transmitted. Even if the thicker metal had the power of transmitting the same degree of temperature by allowing a sufficient lapse of time, yet *time* is an all important element in the case; and evidently the heat abstracted from the metal during the exhaustion of the steam, could not be restored with sufficient rapidity, unless the metal should be thin, and the temperature in the Jacket in excess of that required on the inner walls of the cylinder.

Whether a range of temperature of near 200° can be successfully provided for in one cylinder, without introducing the objectionable effects which result from excessive superheating, and which destroys all the lubricating properties, and causes a great amount of wear,—it is very doubtful. For a given initial pressure, it is quite evident that in the Simple Engine, either the steam will require more superheating, or that the cylinder will require to be raised to a higher

* The range of temperature given on page 310, is 175 degrees. In that view the exhaust line is not taken at the lowest pressure.

temperature by the external application of heat, in order to provide for the greater range of temperature. The balance of experience favours the conclusion that the object under consideration cannot be attained in one cylinder, or, at all events, not without such practical inconveniences as prove fatal to the successful working.

Undoubtedly the subject of STEAM JACKETING would form a more interesting study, and would be much better appreciated, if the laws which govern the operations involved could be thoroughly determined and made apparent. Now the power given out by a Steam Engine is limited by the expenditure of heat; and one of the most vital questions connected with this fact, is the proportion of heat which can be transmuted into useful energy,—that is, into power as understood in connection with the Steam Engine.

Although the whole of any mechanical motion can be converted into heat, yet only a small portion—and usually a very indefinite portion—of heat can be converted into useful motion; so that whilst motion can be converted into heat, it is only partially reconvertible. All the arrangements of the Steam Engine and boiler should be determined with a view of utilizing the largest proportion possible of the heat of the fuel in mechanical motion. Confining our view of this principle in its application to the action of the steam within the cylinder, it may be broadly stated that, to admit steam into the cylinder at the highest practicable temperature, and discharge it into the condenser at the lowest practicable temperature, for a given amount of terminal pressure, will secure the greatest *effective* energy of the steam.

It should always be remembered that all radiation of heat outside the cylinder is absolute loss of power; so that whether the cylinders may be Steam Jacketed or not, they should invariably be so protected or conditioned, as to radiate as little as possible externally. The question of radiation will be treated a little further on.

When steam expands under pressure, or a resisting medium, it not only falls in pressure, but in temperature also. The temperature thus resulting is assumed to correspond to that of steam of the same pressure in the presence of water from which it has been evaporated, and which is called the temperature of evaporation, or boiling point. The heat which has disappeared—assuming here that none has been lost by contact with the metal—has been transmuted into mechanical motion; and some portion of the steam will also be condensed by

reason of its latent heat being subject to similar transmutation; and if special provision be not made for its entire or partial re-evaporation, the cylinder will contain, at the termination of the stroke, steam which carries a considerable admixture of water. The effect of such watery condition of the steam is to absorb a large amount of the heat of the cylinder, and carry it into the condenser, leaving the cylinder cooler by the amount of heat thus abstracted, and so preparing it to condense the next influx of steam.

Bearing on this part of the question, is the fact—of vital importance—that watery steam, or vapour, has an immeasurably greater conducting and absorbing power than dry steam. This has been proved beyond all doubt by the highest scientific experimentalists, and is supported by innumerable proofs and illustrations which are matters of every-day experience. Now, as the steam cannot be maintained in a perfectly dry state during the whole of the time which it remains in the cylinder, then it becomes important to adopt such arrangements as may render this high absorbing power of the steam the least efficacious in acquiring the heat of the metal. When the cylinder is Steam Jacketed, and especially if raised to a sufficiently high temperature, the effect on the vaporous steam within the cylinder appears to be this: The higher temperature of the metal of the cylinder than that of the low pressure steam at the end of the forward stroke, and during exhaustion, will convert the watery particles coming in contact, into steam of slightly higher tension, and which will thus form, in some measure, a protecting medium between the walls of the cylinder and the mass of the steam.

The action here conceived may be illustrated by what takes place when drops of water are let fall upon a hot plate of metal. When the plate is above a certain temperature, the water will continue for some time in the *state of water;* and if examination be made, it will be found that the water is not in actual contact with the metal, and that objects beyond can be seen between the metal and the water. This is the result of a very small portion of the water being converted into steam, the atoms of which are likened, by Professor TYNDALL, to minute projectiles discharged by the hot metal against the water, and thus keeping it apart. When the metal is reduced in temperature, the water will be instantaneously converted into steam because the heat is not sufficient to generate the high projectile force necessary to maintain the separation.

A LAW seems to be involved here, which has not yet been developed. It is more than probable that the separation which will be maintained at any given temperature will be governed by the pressure, and that contact would be instantaneous and complete if the pressure should be sufficiently increased,—no matter how high the temperature might be. And so, if the pressure be reduced, then presumably, the temperature necessary to maintain separation will be lower also. The pressures and temperatures corresponding to the conditions of separation have yet to be determined.

By assuming the sufficiency of the basis for the hypothesis just advanced, we shall be aided in the better understanding of the action of the Steam Jacket. If we suppose the cylinder to be of such a temperature that it will produce the effect which is witnessed in the case of the hot metallic plate and the drops of water, then it is quite evident that the amount of heat abstracted from the metal of the cylinder would not be near so great as when this condition does not obtain. In the case of the drops of water, the heat is transmitted to them *very slowly*, for if otherwise, they would not retain the *state* of water, but would be instantaneously changed into steam. Moreover, the drops of water have been proved, by Professor TYNDALL, by direct experiment, to have only the temperature of water in the liquid state with atmospheric pressure. It seems highly probable that the same principle will operate in the cylinder, and that when the pressure of steam is low, a temperature not excessively high will produce a corresponding effect on the watery steam to that produced on the water by the hot metallic plate. To insure the result here supposed, the temperature of the metal of the cylinder will require to be maintained at a considerable degree. That some such action does take place in the cylinder when Steam Jacketed, is an inference which seems fully warranted by all the known facts connected with the question.

No doubt it will appear paradoxical that, with a higher temperature of the cylinder, the steam issuing from it should carry with it a less quantity of heat. But this is just the result which the Steam Jacket should accomplish ; and if it did not, then the supposed utility of it would be at once disproved most conclusively. As a given weight of steam, of a given pressure and temperature, represents a definite quantity of heat, then, if for a given amount of power, a larger quantity of heat passes into the condenser, so a larger quantity

of steam must have entered the cylinder, because the same amount of heat in each case has been transmuted into mechanical motion. The conclusion at which we have now arrived may perhaps seem to the reader not to harmonise with the laws—as usually understood—of the transmission of heat, in accordance with which it is generally supposed that the loss of heat will be as the difference of temperature. But this law, as we have shown hypothetically, is governed by pressure; and furthermore, that it is greatly modified by the action of the steam in the Jacket, which has the effect of creating a thin atmosphere of dry steam, forming a kind of insulator, or non-conducting medium, between the inner walls of the cylinder and the mass of vaporous steam.

The loss of heat by condensation in the Jackets is small in comparison with the extra heat which, without the Jackets, would be carried away by the steam into the condenser; because, in the Jacket, the steam is not subject to the same conditions which exist within the cylinder. If the quantity of steam condensed within the Jacket was equal to the additional condensation, consequent on loss of heat, which otherwise takes place within the cylinder, then certainly, no benefit would accrue from Steam Jacketing.

In a case of Steam Jacketing of the *low pressure cylinders only* of a pair of Compound Engines, where the steam, after propelling the pistons of the high pressure cylinders, is conducted into the casings of the low pressure cylinders, and thence admitted on the pistons,—the amount of condensation is considerable. Here the conditions are fundamentally different from the cases of Steam Jacketing which we have previously cited. When the steam has given half its available power in the high pressure cylinders—as is the fact in the case referred to,—then, as already shown, it will be in a vaporous state, by reason of the work performed. It is clear then, that after having passed through the transmission pipes, and the casings of the low pressure cylinders, the quantity of water precipitated by condensation must be comparatively great.

Calculations have been given to show the proportion of steam permanently condensed within the cylinders under different ranges of expansion and other varying conditions,—and which condensed steam passes into the condenser in the *state of water*, finely divided, in the form of mist or spray. The reader may perhaps regard such *result, in* whatever degree it may occur, as one to be averted by any

and every means possible. Let us endeavour to find the true value of this view.

In the chapter on Compounding, at page 306, it was stated that with a steam cylinder perfectly non-conducting, and with an initial pressure of 120lbs absolute, and 16 expansions, the condensation by the performance of work would be 0·1953 of the whole. With a less range of expansion the condensation would of course bear a smaller proportion. These results are based on theoretical calculations. In the case of the Compound Engine, represented by Diagrams 42—43, where the actual range of expansion is 14·21, the calculated amount of permanent condensation was found to be 19·61 per cent., and therefore = 0·1961. Here the perfect theoretical efficiency of the steam is well nigh attained, minus the amount of heat abstracted from the steam in the Jackets.

We have now to consider, not alone whether it would be possible to prevent the liquifaction of the steam during expansion, by the external application of heat, or by a high degree of superheating, but whether such result would be advantageous, or productive of economy. As in the case of the Compound Engine just cited (Diagrams 42—43) the permanent condensation is *very little* in excess of what is due to the thermodynamic law, then, clearly, any great reduction in the permanent condensation will necessarily be by an expenditure of heat in excess of that contained in the initial volume of pure steam, proportional to such reduction.

Suppose that it were possible, by enclosing the cylinders with Jackets charged with steam of sufficiently high temperature, &c., and all other conditions such as would re-evaporate all the steam within the cylinder as it became condensed by its primarily contained heat being transmuted into mechanical motion; yet, as exactly the same amount of heat would necessarily be expended in reconverting it into steam, as it had given up in being precipitated, then it is clearly apparent that nothing would be gained by the process. By the facts and hypotheses previously advanced, it may be fairly concluded that no possible amount of heat, supplied by the Steam Jacket, could re-evaporate more than a very small proportion of the condensed steam within the cylinder, except by the lapse of such time as does not obtain in the Steam Engine. For the purpose of superheating the steam it is found necessary to pass it through a congeries of small tubes, amongst which the heated gases circulate, in order that it ma

be broken into small volumes, and so secure a large aggregate area of heated metal with which the steam must inevitably come in contact. The unbroken mass of steam within the cylinder would effectually prevent such rapid communication of heat to it as would insure its re-evaporation. And even if such could be completely accomplished, it would not, as we have already shown, be a gain. Whether or not it would be a loss, so far as obtaining a given average pressure by the expenditure of a given quantity of steam, is not of practical importance, and therefore to pursue such an inquiry could only have an abstract interest, when viewed in relation to the scope of this argument.

Although we have arrived at the conclusion that no gain in economy would arise from the re-evaporation of *that* steam, the condensation of which is governed by the thermodynamic law, it is still important to inquire what will be the practical effect of endeavouring so to re-evaporate the watery particles into steam. It is perfectly certain that such temperature of the metal of the cylinders as could maintain the steam in such purity and dryness as is implied in its leaving the cylinder without any admixture of water, would destroy all the lubrication which might be applied, and would generate so much friction as would prove a prolific source of loss of available power. Condensation of steam within the cylinder being inevitable, it need not be regarded with dismay or regret. What it is important to understand, is the necessary and inevitable amount; and what it is important to avert, is *that* portion of the condensation which results from contact with the metal, and which should be obviated as much as possible, so that the expenditure of heat within the cylinder shall be as little in excess of that which is transmuted into mechanical motion as the best known conditions can insure.

Reverting once more to the case of Diagram 46, and to the calculations and speculations relating thereto, it will be remembered that in the course of this chapter it has been estimated that by compounding, a saving of 25 per cent. would be effected. As the permanent condensation is placed at 33 per cent.; and as that of the Compound Engine (Diagrams 42—43) is placed at 19·61 per cent.; and notwithstanding the statement that the co-efficient of expansion would be greater in the case of Diagram 46 when working compound, the reader may perhaps fail to see how so great a saving can be effected, when so much condensation is required by the law of thermodynamics, as we have pointed out above.

The proportional condensation for 11·4 expansions would obviously be less than for 16 expansions. And further, as on the compound principle the terminal pressure is less for a given initial measure and pressure, so a larger initial measure will necessarily be required to produce the same average pressure of steam throughout the stroke. This cause would reduce the range of expansion as determined by the point of cut-off. The same terminal pressure would yield quite 10 per cent. higher average effective pressure; and as we have shown by analyses and comparison, the permanent condensation, which we found to be 33 per cent. of the whole consumption, would be reduced by 50 per cent., leaving 16·5 per cent. as the proportion of permanent condensation when working on the compound principle. This amount, for the range of expansion just supposed, would be approximately proportional to 19·61 per cent. for 14·21 expansions in the Compound Engine, Diagrams 42—43. If, on changing to the compound principle, the present terminal pressure and range of expansion should be maintained, then the initial pressure would require to be about 90lbs absolute. This would increase the co-efficient of expansion still more. The increased co-efficient of expansion referred to on page 327, is on the assumption of the same initial measure and pressure of steam. The probable proportion of permanent condensation which would obtain in the case of Diagram 46, when compounded, and which is given just above at 16·5 per cent., would require some small portion of the steam which had been condensed by the performance of work, to be re-evaporated.

The consumption of steam by condensation in the Jackets as hitherto working on the *Simple* principle, amounts to 3·4 per cent. of the whole. The expenditure of heat represented by this proportion would be quite sufficient, when working on the *Compound* principle, to insure so much re-evaporation of the steam necessarily condensed, that the permanent condensation would not exceed 16·5 per cent., as already stated. Add to this the gain by the higher co-efficient of expansion, and a gain of 25 per cent. by compounding may be confidently relied upon.

A few words more—and they shall be final—on the co-efficient of expansion. The principle propounded in the chapter on Compounding, at pages 304—305, is based upon the assumption that the *weight* of steam at the point of cut-off—that is, the initial measure —bears a *constant* proportion to such initial measure and pressure

On this assumption, the terminal pressure, for a given initial pressure and range of expansion, would be above or below a given line—say the isothermal—according as the steam may be more re-evaporated or condensed progressively during its expansion.

If we consider the principle from the point of view of the terminal pressure only ; and if in two cases we find, on investigation, that for the same terminal pressure, one has a higher average effective pressure by 10 per cent., and yet its steam at the terminal pressure is of equal purity, that is, of no greater degree of saturation, then, beyond doubt, there will be a clear gain of 10 per cent. Any further reduction in the degree of saturation at the terminal pressure, in the case of the higher co-efficient, will obviously be accompanied by an equivalent gain in economy, as fully shown in discussing the question of permanent condensation.

Apart from the question of temperature, the conditions of the sides and ends of the cylinders are not alike ; for whilst the sides are bright and smooth,—exposing to the action of the steam a clean metallic surface,—the ends are invariably black, dirty, and rough on the surface. This is important to be borne in mind, as it leads to another phase of the subject, which, though not appertaining *solely* to the question of Steam Jacketing, may be most appropriately introduced in this connection.

A great variety of refined and exact experiments have conclusively proved that radiation and absorption of heat by iron—as well as other metals—is small when the surface is clean and polished. It is indeed exceedingly small. When the surface is coated with any heterogeneous or compound substance, its radiating or absorbing power is immensely increased.* Now let us consider this fact in its relation to the internal condition of the cylinder of the Steam Engine. If the direct radiation and absorption of heat by the internal surfaces of the cylinder can be prevented, it will be obvious that one of the disturbing causes within the cylinder will have been removed. Of the three modes of the transmission of heat, two—conduction and convection—have already been treated. Accepting the law, as stated above, of the non-radiating and non-absorbing properties of polished metallic surfaces, it is evident that the inner surfaces of the cylinder covers, and the two faces of the piston, ought to be protected by

* Let the reader test the radiation of a bright and pure metallic tea-pot, and of a black kettle, each containing water of the same temperature, and he will find a very sensible difference.

polished surfaces. It seems clear that if these conditions existed, little heat would be lost by internal radiation.

That a much greater ratio of gain by Steam Jacketing does not obtain in the case of the Engines represented by Diagram 46 than in the case of Diagrams 42—43, may possibly, in some measure, be due to excessive radiation from the covers of the cylinders. As vaporous steam has the property of absorbing radiant heat in a very high degree, this view of the case derives considerable force. And the higher the temperature to which the radiating substance may be raised, the greater will be the amount of heat radiated. The absorption of the radiant heat by the steam being greater as it is more heavily charged with water, then the purer and dryer the steam may be, the smaller will be the amount of heat thus absorbed,—all other conditions being equal. As steam will invariably become vaporous, or watery, in the cylinder; and as in every condition it will absorb radiant heat more or less; so it will be apparent that this is a circumstance not altogether so trifling as to be undeserving of consideration.

Let the reader stand by the cylinder of a Steam Engine which is black and dirty on every part of its external surface, and which is charged with steam of, say 60lbs pressure, and endeavour to estimate the amount of heat radiated externally; then conceive the whole of such radiant heat to be swept entirely away at every revolution of the Engine, and the loss thus occurring will begin to be apprehended. Such is the nature of the operation proceeding *within the cylinder*, just in proportion to the extent of the radiating area, and its condition as compared with that just conceived to exist outside. When the facts and conditions which have been here surveyed have been fully contemplated, it will be seen that radiation may play an important part in the economy of the Steam Engine.

If, instead of Iron, a metal of less specific heat, and lower conductivity, could be made to serve for the cylinder and covers and piston, a decided advantage would be obtained thereby. There does not seem at present any possibility of adopting a material which will be entirely non-conducting. Nor is there the prospect of superseding Iron as the material for the working, or frictional part of the cylinder: that is, Iron in some of its *states*, as *cast*, *wrought*, and *steel*. The faces of the piston, and the inner faces of the covers, however, are not limited to one metal by the same indispensable conditions.

If the piston and the internal surfaces of the cylinder covers could be faced with the metal BISMUTH, it is more than probable that considerable economy would result, because, as compared with iron, its specific heat is little more than one-fourth, and its conductivity only one-sixth. This being done, Steam Jacketing of the cylinder covers would probably be superfluous. Indeed, the loss would be more effectually obviated than by Steam Jacketing, when the inner faces of the covers are not kept clean and bright.

The importance of supplying steam to the Jackets of a high temperature has been urged, and the question of superheating has been introduced, and its danger incidentally referred to. It is well known that superheating the steam before it enters the cylinder secures economy by the increased power of a given quantity of steam. But highly superheated steam, passing through the valves of the cylinder, is productive of much mischief, inasmuch as it robs all the rubbing surfaces of all lubrication, and thereby causes great wearing of the moving parts, excessive friction, and ultimate leakage,—sometimes to a serious extent. A very moderate superheating of the steam before entering the cylinder would be useful; but the beneficial extent of such superheating must be determined in each particular case by experience.

In the case of a COMPOUND ENGINE, in which the cylinders are Steam Jacketed, the superheating of the steam before entering the high pressure cylinder might be objectionable, because of the smaller range in the alternations of temperature to which it is subjected; if superheated at all, it should only be so much as will prevent any more condensation by contact with the surface of the metal than is just requisite for the proper and useful lubrication; but it would be advantageous to give the steam a little superheating in the RECEIVER, or in passing from the high to the low pressure cylinder, as the steam here, being of a lower pressure and temperature, would not be so liable to injure the rubbing surfaces. And besides, the steam having already performed work, and having, further, to come next in contact with the internal surfaces of the low pressure cylinder, which have just been exposed to the cooling influence of the condenser, will obviously admit of a greater quantity of heat being imparted to it without tending to any objectionable effect.

The useless expenditure of heat in every way should be carefully *guarded* against, in order to secure the most economical results. It

would be possible to have the cylinder separated—that is by insulation —from the bed plate, or other foundation to which it may be secured, by a low conducting material; and also to have the cylinder, and such other parts as are subject to a high temperature, made almost non-radiating. These conditions would be found advantageous in preventing the useless diffusion of heat; and also in keeping the Engine-house at a comfortable temperature.

To prevent the escape of heat by radiation, it is only necessary that the cylinder, and all parts of the metal which receive heat from the steam by contact or conduction, should have a clean metallic surface. Whoever has been much about Steam Engines will have often felt—and many without having been conscious of the cause—that a dirty cylinder, and Engine generally, seemed to make the Engine-room insufferably hot. Such is really the fact. If anyone should doubt the statement that a clean polished metallic surface radiates less heat than a black and dirty one, let him try the experiment, and he will doubt no more. This is one of the facts in connection with the laws of the transmission of heat, which experimental science has completely established. Here, then, we see, that cleanliness is the handmaid of economy, and, at the same time, pleasant withal to look upon.

W. S.

APPENDIX:

CONTAINING VARIOUS

TABLES, &c.,

USEFUL AND INTERESTING TO THE

ENGINEER.

A TABLE FOR SAFETY VALVES, CYLINDERS, AIR-PUMPS, &c.,

Containing the Circumferences and Areas of Circles from $\frac{1}{16}$ of an inch to 10 inches, advancing by $\frac{1}{16}$ of an inch; and by $\frac{1}{8}$ of an inch, from 10 inches to 100 inches Diameter.

Diameter.	Circum.	Area.	Diameter.	Circum.	Area.
$\frac{1}{16}$	·1963	·0030	2 inch.	6·2832	3·1416
$\frac{1}{8}$ ...	·3927	·0122	$\frac{1}{16}$	6·4795	3·3411
$\frac{3}{16}$	·5890	·0276	$\frac{1}{8}$ ...	6·6759	3·5465
$\frac{1}{4}$ ...	·7854	·0490	$\frac{3}{16}$	6·8722	3·7582
$\frac{5}{16}$	·9817	·0767	$\frac{1}{4}$ ...	7·0686	3·9760
$\frac{3}{8}$ ...	1·1781	·1104	$\frac{5}{16}$	7·2649	4·2001
$\frac{7}{16}$	1·3744	·1503	$\frac{3}{8}$ ...	7·4613	4·4302
$\frac{1}{2}$ ...	1·5708	·1963	$\frac{7}{16}$	7·6576	4·6664
$\frac{9}{16}$	1·7671	·2485	$\frac{1}{2}$ ...	7·8540	4·9087
$\frac{5}{8}$ ...	1·9635	·3068	$\frac{9}{16}$	8·0503	5·1573
$\frac{11}{16}$	2·1598	·3712	$\frac{5}{8}$ ...	8·2467	5·4119
$\frac{3}{4}$ ...	2·3562	·4417	$\frac{11}{16}$	8·4430	5·6727
$\frac{13}{16}$	2·5525	·5185	$\frac{3}{4}$ ...	8·6394	5·9395
$\frac{7}{8}$ ...	2·7489	·6013	$\frac{13}{16}$	8·8357	6·2126
$\frac{15}{16}$	2·9452	·6903	$\frac{7}{8}$ ...	9·0321	6·4918
			$\frac{15}{16}$	9·2284	6·7772
1 inch.	3·1416	·7854			
$\frac{1}{16}$	3·3379	·8861	3 inch.	9·4248	7·0686
$\frac{1}{8}$ ...	3·5343	·9940	$\frac{1}{16}$	9·6211	7·3662
$\frac{3}{16}$	3·7306	1·1075	$\frac{1}{8}$ ...	9·8175	7·6699
$\frac{1}{4}$ ...	3·9270	1·2271	$\frac{3}{16}$	10·0138	7·9798
$\frac{5}{16}$	4·1233	1·3529	$\frac{1}{4}$ ...	10·2102	8·2957
$\frac{3}{8}$ ...	4·3197	1·4848	$\frac{5}{16}$	10·4065	8·6179
$\frac{7}{16}$	4·5160	1·6229	$\frac{3}{8}$ ...	10·6029	8·9462
$\frac{1}{2}$ ...	4·7124	1·7671	$\frac{7}{16}$	10·7992	9·2806
$\frac{9}{16}$	4·9087	1·9175	$\frac{1}{2}$ ...	10·9956	9·6211
$\frac{5}{8}$ ...	5·1051	2·0739	$\frac{9}{16}$	11·1919	9·9678
$\frac{11}{16}$	5·3014	2·2365	$\frac{5}{8}$ ...	11·3883	10·3206
$\frac{3}{4}$ ...	5·4978	2·4052	$\frac{11}{16}$	11·5846	10·6796
$\frac{13}{16}$	5·6941	2·5801	$\frac{3}{4}$ ...	11·7810	11·0446
$\frac{7}{8}$ ...	5·8905	2·7611	$\frac{13}{16}$	11·9773	11·4159
$\frac{15}{16}$	6·0868	2·9483	$\frac{7}{8}$ ...	12·1737	11·7932

Diameter.	Circum.	Area.	Diameter.	Circum.	Area.
$\frac{15}{16}$	12·3700	12·1768	$\frac{1}{2}$ ...	20·4204	33·1831
			$\frac{9}{16}$	20·6167	33·8244
4 inch.	12·5664	12·5664	$\frac{5}{8}$ ...	20·8131	34·4717
$\frac{1}{16}$	12·7627	12·9622	$\frac{11}{16}$	21·0094	35·1252
$\frac{1}{8}$ ...	12·9591	13·3640	$\frac{3}{4}$ ...	21·2058	35·7847
$\frac{3}{16}$	13·1554	13·7721	$\frac{13}{16}$	21·4021	6·4505
$\frac{1}{4}$ ...	13·3518	14·1862	$\frac{7}{8}$ ...	21·5985	37·1224
$\frac{5}{16}$	13·5481	14·6066	$\frac{15}{16}$	21·7948	37·8005
$\frac{3}{8}$ ...	13·7445	15·0331			
$\frac{7}{16}$	13·9408	15·4657	7 inch.	21·9912	38·4846
$\frac{1}{2}$ ...	14·1372	15·9043	$\frac{1}{16}$	22·1875	39·1749
$\frac{9}{16}$	14·3335	16·3492	$\frac{1}{8}$ ...	22·3839	39·8713
$\frac{5}{8}$ ...	14·5299	16·8001	$\frac{3}{16}$	22·5802	40·5469
$\frac{11}{16}$	14·7262	17·2573	$\frac{1}{4}$ ...	22·7766	41·2825
$\frac{3}{4}$ ...	14·9226	17·7205	$\frac{5}{16}$	22·9729	41·9974
$\frac{13}{16}$	15·1189	18·1900	$\frac{3}{8}$ ...	23·1693	42·7184
$\frac{7}{8}$ ...	15·3153	18·6655	$\frac{7}{16}$	23·3656	43·4455
$\frac{15}{16}$	15·5716	19·1472	$\frac{1}{2}$ ...	23·5620	44·1787
			$\frac{9}{16}$	23·7583	44·9181
5 inch.	15·7080	19·6350	$\frac{5}{8}$ ...	23·9547	45·6636
$\frac{1}{16}$	15·9043	20.1290	$\frac{11}{16}$	24·1510	46·4153
$\frac{1}{8}$ ...	16·1007	20·6290	$\frac{3}{4}$ ...	24·3474	47·1730
$\frac{3}{16}$	16·2970	21·1252	$\frac{13}{16}$	24·5437	47·9370
$\frac{1}{4}$ ...	16·4934	21·6475	$\frac{7}{8}$ ...	24·7401	48·7070
$\frac{5}{16}$	16·6897	22·1661	$\frac{15}{16}$	24·9364	49·4833
$\frac{3}{8}$ ...	16·8861	22·6907			
$\frac{7}{16}$	17·0824	23·2215	8 inch.	25·1328	50·2656
$\frac{1}{2}$ ...	17·2788	23·7583	$\frac{1}{16}$	25·3291	51·0541
$\frac{9}{16}$	17·4751	24·3014	$\frac{1}{8}$ ...	25·5255	51·8486
$\frac{5}{8}$ ...	17·6715	24·8505	$\frac{3}{16}$	25·7218	52·8994
$\frac{11}{16}$	17·8678	25·4058	$\frac{1}{4}$ ...	25·9182	53·4562
$\frac{3}{4}$ ...	18·0642	25·9672	$\frac{5}{16}$	26·1145	54·2748
$\frac{13}{16}$	18·2605	26·5348	$\frac{3}{8}$ ...	26·3109	55·0885
$\frac{7}{8}$ ...	18·4569	27·1085	$\frac{7}{16}$	26·5072	55·9138
$\frac{15}{16}$	18·6532	27·6884	$\frac{1}{2}$ ...	26·7036	56·7451
			$\frac{9}{16}$	26·8999	57·5887
6 inch.	18·8496	28·2744	$\frac{5}{8}$ ...	27·0963	58·4264
$\frac{1}{16}$	19·0459	28·8665	$\frac{11}{16}$	27·2926	59·7762
$\frac{1}{8}$ ...	19·2423	29·4647	$\frac{3}{4}$ ...	27·4890	60·1321
$\frac{3}{16}$	19·4386	30·0798	$\frac{13}{16}$	27·6853	60·9943
$\frac{1}{4}$ ...	19·6350	30·6796	$\frac{7}{8}$ ...	27·8817	61·8625
$\frac{5}{16}$	19·8313	31·2964	$\frac{15}{16}$	27·0780	62·7369
$\frac{3}{8}$ ...	20·0277	31·9192			
$\frac{7}{16}$	20·2240	32·5481	9 inch.	28·2744	63·6174

Diameter.	Circum.	Area.
$\frac{1}{16}$	28·4707	64·5041
$\frac{1}{8}$ ...	28·6671	65·3968
$\frac{3}{16}$	28·8634	66·2957
$\frac{1}{4}$ ...	29·0598	67·2007
$\frac{5}{16}$	29·2561	68·1120
$\frac{3}{8}$ ...	29·4525	69·0293
$\frac{7}{16}$	29·6488	69·9528
$\frac{1}{2}$ ...	29·8452	70·8823
$\frac{9}{16}$	30·0415	71·8181
$\frac{5}{8}$ ...	30·2379	72·7599
$\frac{11}{16}$	30·4342	73·7079
$\frac{3}{4}$ ...	30·6306	74·6620
$\frac{13}{16}$	30·8269	75·6223
$\frac{7}{8}$ ...	31·0233	76·5887
$\frac{15}{16}$	31·2196	77·5613
10 inch.	31·4160	78·5400
$\frac{1}{8}$	31·8087	80·5157
$\frac{1}{4}$ ...	32·2014	82·5160
$\frac{3}{8}$	32·5941	84·5409
$\frac{1}{2}$ ...	32·9868	86·5903
$\frac{5}{8}$	33·3795	88·6643
$\frac{3}{4}$ ...	33·7722	90·7627
$\frac{7}{8}$	34·1649	92·8858
11 inch.	34·5576	95·0334
$\frac{1}{8}$	34·9503	97·2053
$\frac{1}{4}$ ...	35·3430	99·4021
$\frac{3}{8}$	35·7357	101·6234
$\frac{1}{2}$ ...	36·1284	103·8691
$\frac{5}{8}$	36·5211	106·1394
$\frac{3}{4}$ ...	36·9138	108·4342
$\frac{7}{8}$	37·3065	110·7536
12 inch.	37·6992	113·0976
$\frac{1}{8}$	38·0919	115·4660
$\frac{1}{4}$ ...	38·4846	117·8590
$\frac{3}{8}$	38·8773	120·2766
$\frac{1}{2}$ ...	39·2700	122·7187
$\frac{5}{8}$	39·6627	125·1854
$\frac{3}{4}$ ...	40·0554	127·6765
$\frac{7}{8}$	40·4481	130·1923
13 inch.	43·8408	132·7326

Diameter.	Circum.	Area.
$\frac{1}{8}$	41·2338	135·2974
$\frac{1}{4}$ ...	41·6262	137·8867
$\frac{3}{8}$	42·0189	140·5007
$\frac{1}{2}$ ..	42·4116	143·1391
$\frac{5}{8}$	42·8043	145·8021
$\frac{3}{4}$ ...	43·1970	148·4896
$\frac{7}{8}$	43·5897	151·2017
14 inch.	43·9824	153·9384
$\frac{1}{8}$	44·3751	156·6995
$\frac{1}{4}$ ...	44·7676	159·4852
$\frac{3}{8}$	45·1605	162·2956
$\frac{1}{2}$ ...	45·5532	165·1303
$\frac{5}{8}$	45·9459	167·9896
$\frac{3}{4}$ ...	46·3386	170·8735
$\frac{7}{8}$	46·7313	173·7820
15 inch.	47·1240	176·7150
$\frac{1}{8}$	47·5167	179·6725
$\frac{1}{4}$ ...	47·9094	182·6545
$\frac{3}{8}$	48·3021	185·6612
$\frac{1}{2}$ ...	48·6948	188·6923
$\frac{5}{8}$	49·0875	191·7480
$\frac{3}{4}$ ...	49·4802	194·8282
$\frac{7}{8}$	49·8729	197·9330
16 inch.	50·2656	201·0624
$\frac{1}{8}$	50·6583	204·2162
$\frac{1}{4}$ ...	51·0510	207·3946
$\frac{3}{8}$	51·4437	210·5976
$\frac{1}{2}$ ...	51·8364	213·8251
$\frac{5}{8}$	52·2291	217·0772
$\frac{3}{4}$ ...	52·6218	220·3537
$\frac{7}{8}$	53·0145	223·6549
17 inch.	53·4072	226·9806
$\frac{1}{8}$	53·7999	230·3308
$\frac{1}{4}$ ...	54·1926	233·7055
$\frac{3}{8}$	54·5853	237·1049
$\frac{1}{2}$ ...	54·9780	240·5287
$\frac{5}{8}$	55·3707	243·9771
$\frac{3}{4}$ ...	55·7634	247·4500
$\frac{7}{8}$	56·1561	250·9475

Diam.	Circum.	Area.	Diam.	Circum.	Area.
18 inch	56·5488	254·4696	23 inch	72·2568	415·4766
1/8	56·9415	258·0161	1/8	72·6495	420·0049
1/4 ...	57·3342	261·5872	1/4 ...	73·0422	424·5577
3/8	57·7269	265·1829	3/8	73·4349	429·1352
1/2 ...	58·1196	268·8031	1/2 ...	73·8276	433·7371
5/8	58·5123	272·4479	5/8	74·2203	438·3636
3/4 ...	58·9056	276·1171	3/4 ...	74·6130	443·0146
7/8	59·2977	279·8110	7/8	75·0057	447·6992
19 inch	59·6904	283·5294	24 inch	75·3984	452·3904
1/8	60·0831	287·2723	1/8	75·7911	457·1150
1/4 ...	60·4758	291·0397	1/4 ...	76·1838	461·8642
3/8	60·8685	294·8312	3/8	76·5765	466·6380
1/2 ...	61·2612	298·6483	1/2 ...	76·9692	471·4363
5/8	61·6539	302·4894	5/8	77·3619	476·2592
3/4 ...	62·0466	306·3550	3/4 ...	77·7546	481·1065
7/8	62·4393	310·2452	7/8	78·1473	485·9785
20 inch	62·8320	314·1600	25 inch	78·5400	490·8750
1/8	63·2247	318·0992	1/8	78·9327	495·7960
1/4 ...	63·6174	322·0630	1/4 ...	79·3254	500·7415
3/8	64·0101	326·0514	3/8	79·7181	505·7117
1/2 ...	64·4028	330·0643	1/2 ...	80·1108	510·7063
5/8	64·7955	334·1018	5/8	80·5035	515·7255
3/4 ...	65·1882	338·1637	3/4 ...	80·8962	520·7692
7/8	65·5809	342·2503	7/8	81·2889	525·8375
21 inch	65·7936	346·3614	26 inch	81·6816	530·9304
1/8	66·3663	350·4970	1/8	82·0743	536·0477
1/4 ...	66·7590	354·6571	1/4 ...	82·4670	541·1896
3/8	67·1517	358·8419	3/8	82·8597	546·3561
1/2 ...	67·5444	363·0511	1/2 ...	83·2524	551·5471
5/8	67·9371	367·2849	5/8	83·6451	556·7627
3/4 ...	68·3298	371·5432	3/4 ...	84·0378	562·0027
7/8	68·7225	375·8261	7/8	84·4305	567·2674
22 inch	69·1152	380·1336	27 inch	84·8232	572·5566
1/8	69·5079	384·4655	1/8	85·2159	577·8703
1/4 ...	69·9006	388·8220	1/4 ...	85·6086	583·2085
3/8	70·2933	393·2031	3/8	86·0013	588·5714
1/2 ...	70·6860	397·6087	1/2 ...	86·3940	593·9587
5/8	71·0787	402·0388	5/8	86·7867	599·3706
3/4 ..	71·4714	406·4935	3/4 ...	87·1794	604·8070
7/8	71·8641	410·9728	7/8	87·5721	610·2680

Diam.	Circum.	Area.	Diam.	Circum.	Area.
28 inch	87·9648	615·7536	33 inch	103·6728	855·3006
⅛	88·3575	621·2636	⅛	104·0655	861·7924
¼ ...	88·7502	626·7932	¼ ...	104·4582	868·3087
⅜	89·1429	632·3574	⅜	104·8509	874·8497
½ ...	89·5356	637·9411	½ ...	105·2436	881·4151
⅝	89·9283	643·5494	⅝	105·6363	888·0051
¾ ...	90·3210	649·1821	¾ ...	106·0290	894·6196
⅞	90·7137	654·8395	⅞	106·4217	901·2587
29 inch	91·1064	660·5214	34 inch	106·8144	907·9224
⅛	91·4991	666·2278	⅛	107·2071	914·6105
¼ ...	91·8918	671·9587	¼ ...	107·5998	921·3232
⅜	92·2845	677·7143	⅜	107·9925	928·0605
½ ...	92·6772	683·4943	½ ...	108·3852	934·8223
⅝	93·0699	689·2989	⅝	108·7779	941·6086
¾ ...	93·4626	695·1280	¾ ...	109·1706	948·4195
⅞	93·8553	700·9817	⅞	109·5633	955·2550
30 inch	94·2480	706·8600	35 inch	109·9560	962·1150
⅛	94·6407	712·7627	⅛	110·3487	968·9995
¼ ...	95·0334	718·6900	¼ ...	110·7414	975·9085
⅜	95·4261	724·6419	⅜	111·1341	982·8422
½ ...	95·8188	730·6183	½ ...	111·5268	989·8003
⅝	96·2115	736·6193	⅝	111·9195	996·7830
¾ ...	96·6042	742·6447	¾ ...	112·3122	1003·7902
⅞	96·9969	748·6948	⅞	112·7049	1010·8220
31 inch	97·3896	754·7694	36 inch	113·0976	1017·8784
⅛	97·7823	760·8685	⅛	113·4903	1024·9592
¼ ...	98·1750	766·9921	¼ ...	113·8830	1032·0646
⅜	98·5677	773·1404	⅜	114·2757	1039·1946
½ ...	98·9684	779·3131	½ ...	114·6684	1046·3941
⅝	99·3531	785·5104	⅝	115·0611	1053·5281
¾ ...	99·7458	791·7322	¾ ...	115·4538	1060·7317
⅞	100·1385	797·9786	⅞	115·8465	1067·9599
32 inch	100·5312	804·2496	37 inch	116·2392	1075·2126
⅛	100·9240	810·5450	⅛	116·6319	1082·4898
¼ ...	101·3166	816·8650	¼ ...	117·0246	1089·7915
⅜	101·7093	823·2096	⅜	117·4173	1097·1179
½ ...	102·1020	829·5787	½ ...	117·8100	1104·4687
⅝	102·4947	835·9724	⅝	118·2027	1111·8441
¾ ...	102·8874	842·3905	¾ ...	118·5954	1119·2440
⅞	103·2801	848·8333	⅞	118·9881	1126·6685

Diam.	Circum.	Area.	Diam.	Circum.	Area.
38 inch	119·3808	1134·1176	43 inch	135·0888	1452·2046
$\frac{1}{8}$	119·7735	1141·5911	$\frac{1}{8}$	135·4815	1460·6599
$\frac{1}{4}$ ...	120·1662	1149·0892	$\frac{1}{4}$ ...	135·8742	1469·1397
$\frac{3}{8}$	120·5589	1156·6119	$\frac{3}{8}$	136·2669	1477·6342
$\frac{1}{2}$ ...	120·9516	1164·1591	$\frac{1}{2}$ ..	136·6596	1486·1751
$\frac{5}{8}$	121·3443	1171·7309	$\frac{5}{8}$	137·0523	1494·7266
$\frac{3}{4}$ ...	121·7370	1179·3271	$\frac{3}{4}$ ..	137·4450	1503·3046
$\frac{7}{8}$	122·1297	1186·9480	$\frac{7}{8}$	137·8377	1511·9072
39 inch	122·5224	1194·5934	44 inch	138·2304	1520·5344
$\frac{1}{8}$	122·9151	1202·2633	$\frac{1}{8}$	138·6231	1529·1860
$\frac{1}{4}$ ...	123·3078	1209·9577	$\frac{1}{4}$ ...	139·0158	1537·8622
$\frac{3}{8}$	123·7005	1217·6768	$\frac{3}{8}$	139·4085	1546·5530
$\frac{1}{2}$ ...	124·0932	1225·4203	$\frac{1}{2}$ ...	139·8012	1555·2883
$\frac{5}{8}$	124·4859	1233·1884	$\frac{5}{8}$	140·1939	1564·0382
$\frac{3}{4}$ ...	124·9787	1240·9810	$\frac{3}{4}$ ...	140·5866	1572·8125
$\frac{7}{8}$	125·2713	1248·7982	$\frac{7}{8}$	140·9793	1581·6115
40 inch	125·6640	1256·6400	45 inch	141·3720	1590·4350
$\frac{1}{8}$	126·0567	1264·5062	$\frac{1}{8}$	141·7647	1599·2830
$\frac{1}{4}$ ...	126·4494	1272·3970	$\frac{1}{4}$ ...	142·1574	1608·1555
$\frac{3}{8}$	126·8421	1280·3124	$\frac{3}{8}$	142·5501	1617·0427
$\frac{1}{2}$ ...	127·2348	1288·2523	$\frac{1}{2}$ ...	142·9428	1625·9743
$\frac{5}{8}$	127·6275	1296·2168	$\frac{5}{8}$	143·3355	1634·9205
$\frac{3}{4}$ ...	128·0202	1304·2057	$\frac{3}{4}$ ...	143·7382	1643·8912
$\frac{7}{8}$	128·4129	1312·2193	$\frac{7}{8}$	144·1209	1652·8865
41 inch	128·8056	1320·2574	46 inch	144·5136	1661·9064
$\frac{1}{8}$	129·1983	1328·3200	$\frac{1}{8}$	144·9063	1670·9507
$\frac{1}{4}$ ...	129·5910	1336·4071	$\frac{1}{4}$ ...	145·2990	1680·0196
$\frac{3}{8}$	129·9837	1344·5189	$\frac{3}{8}$	145·6917	1689·1031
$\frac{1}{2}$ ...	130·3764	1352·6551	$\frac{1}{2}$ ...	146·0844	1698·2311
$\frac{5}{8}$	130·7691	1360·8159	$\frac{5}{8}$	146·4771	1707·3737
$\frac{3}{4}$ ...	131·1618	1369·0012	$\frac{3}{4}$ ...	146·8698	1716·5407
$\frac{7}{8}$	131·5545	1377·2111	$\frac{7}{8}$	147·2625	1725·7324
42 inch	131·9472	1385·4456	47 inch	147·6552	1734·9486
$\frac{1}{8}$	132·3399	1393·7045	$\frac{1}{8}$	148·0479	1744·1893
$\frac{1}{4}$ ...	132·7326	1401·9880	$\frac{1}{4}$ ...	148·4406	1753·4545
$\frac{3}{8}$	133·1253	1470·2961	$\frac{3}{8}$	148·8333	1762·7344
$\frac{1}{2}$ ...	133·5180	1418·6287	$\frac{1}{2}$ ...	149·2260	1772·0587
$\frac{5}{8}$	133·9107	1426·9859	$\frac{5}{8}$	149·6187	1781·3976
$\frac{3}{4}$ ...	134·3034	1435·3675	$\frac{3}{4}$ ...	150·0114	1790·7610
$\frac{7}{8}$	134·6961	1443·7738	$\frac{7}{8}$	150·4041	1800·1490

Diam.	Circum.	Area.	Diam.	Circum.	Area.
48 inch	150·7968	1809·5616	53 inch	166·5048	2206·1886
$\frac{1}{8}$	151·1895	1818·9986	$\frac{1}{8}$	166·8975	2216·6074
$\frac{1}{4}$ ...	151·5822	1828·4602	$\frac{1}{4}$ ...	167·2902	2227·0507
$\frac{3}{8}$	151·9749	1837·9364	$\frac{3}{8}$	167·6829	2237·5187
$\frac{1}{2}$ ...	152·3676	1847·4571	$\frac{1}{2}$ ...	168·0756	2248·0111
$\frac{5}{8}$	152·7603	1856·9924	$\frac{5}{8}$	168·4683	2258·5281
$\frac{3}{4}$ ...	153·1530	1868·5521	$\frac{3}{4}$ ...	168·8610	2269·0696
$\frac{7}{8}$	153·5457	1876·1365	$\frac{7}{8}$	169·2537	2279·6357
49 inch	153·9384	1885·7454	54 inch	169·6464	2290·2264
$\frac{1}{8}$	154·3311	1895·3788	$\frac{1}{8}$	170·0391	2300·8415
$\frac{1}{4}$ ...	154·7238	1905·0367	$\frac{1}{4}$ ...	170·4318	2311·4812
$\frac{3}{8}$	155·1165	1914·7093	$\frac{3}{8}$	170·8245	2322·1455
$\frac{1}{2}$ ...	155·5092	1924·4263	$\frac{1}{2}$ ...	171·2172	2332·8343
$\frac{5}{8}$	155·9019	1934·1579	$\frac{5}{8}$	171·6099	2343·5477
$\frac{3}{4}$ ...	156·2946	1943·9140	$\frac{3}{4}$ ...	172·0026	2354·2855
$\frac{7}{8}$	156·6873	1953·6947	$\frac{7}{8}$	172·3593	2365·0480
50 inch	157·0800	1963·5000	55 inch	172·7880	2375·8350
$\frac{1}{8}$	157·4727	1973·3297	$\frac{1}{8}$	173·1807	2386·6465
$\frac{1}{4}$ ...	157·8654	1983·1840	$\frac{1}{4}$ ...	173·5734	2397·4825
$\frac{3}{8}$	158·2581	1993·0529	$\frac{3}{8}$	173·9661	2408·3432
$\frac{1}{2}$ ...	158·6508	2002·9663	$\frac{1}{2}$ ...	174·3588	2419·2283
$\frac{5}{8}$	159·0435	2012·8943	$\frac{5}{8}$	174·7515	2430·1833
$\frac{3}{4}$ ...	159·4362	2022·8467	$\frac{3}{4}$ ...	175·1442	2441·0772
$\frac{7}{8}$	159·8289	2032·8238	$\frac{7}{8}$	175·5369	2452·0310
51 inch	160·2216	2042·8254	56 inch	175·9296	2463·0144
$\frac{1}{8}$	160·6143	2052·8515	$\frac{1}{8}$	176·3323	2474·0222
$\frac{1}{4}$ ...	161·0070	2062·9021	$\frac{1}{4}$ ...	176·7150	2485·3546
$\frac{3}{8}$	161·3997	2072·9764	$\frac{3}{8}$	177·1077	2496·1116
$\frac{1}{2}$ ...	161·7924	2083·0771	$\frac{1}{2}$ ...	177·5004	2507·1931
$\frac{5}{8}$	162·1851	2093·2014	$\frac{5}{8}$	177·8931	2518·2992
$\frac{3}{4}$ ...	162·5778	2103·3502	$\frac{3}{4}$ ...	178·2858	2529·4297
$\frac{7}{8}$	162·9705	2113·5236	$\frac{7}{8}$	178·6785	2543·5849
52 inch	163·3632	2123·7216	57 inch	179·0712	2551·7646
$\frac{1}{8}$	163·7559	2133·9440	$\frac{1}{8}$	179·4639	2562·9688
$\frac{1}{4}$ ..	164·1486	2144·1910	$\frac{1}{4}$ ..	179·8566	2574·1975
$\frac{3}{8}$	164·5413	2154·4626	$\frac{3}{8}$	180·2493	2585·4509
$\frac{1}{2}$ ...	164·9340	2164·7587	$\frac{1}{2}$ ...	180·6423	2596·7287
$\frac{5}{8}$	165·3267	2175·0794	$\frac{5}{8}$	181·0347	2608·0311
$\frac{3}{4}$ ..	165·7194	2185·4245	$\frac{3}{4}$ ...	181·4274	2619·3580
$\frac{7}{8}$	166·1121	2195·7943	$\frac{7}{8}$	181·8201	2630·7095

Diam.	Circum.	Area.	Diam.	Circum.	Area.
58 inch	182·2128	2642·0856	63 inch	197·9208	3117·2526
⅛	182·6055	2653·4861	⅛	198·3135	3129·6349
¼ ...	182·9982	2664·9112	¼ ...	198·7062	3142·0417
⅜	183·3909	2676·3609	⅜	199·0989	3154·4732
½ ...	183·7836	2687·8551	½ ...	199·4916	3166·9291
⅝	184·1763	2699·3338	⅝	199·8843	3179·4096
¾ ...	184·5690	2710·8571	¾ ..	200·2770	3191·9146
⅞	184·9617	2722·4050	⅞	200·6697	3204·4442
59 inch	185·3544	2733·9774	64 inch	201·0624	3216·9984
⅛	185·7471	2745·5743	⅛	201·4551	3229·5770
¼ ...	186·1398	2757·1957	¼ ...	201·8478	3242·1782
⅜	186·5325	2768·8418	⅜	202·2405	3254·8080
½ ...	186·9252	2780·5123	½ ...	202·6332	3267·4603
⅝	187·3179	2792·2074	⅝	203·0259	3280·1372
¾ ...	187·7106	2803·9270	¾ ..	203·4186	3292·8385
⅞	188·1033	2815·6712	⅞	203·8113	3305·5645
60 inch	188·4960	2827·4400	65 inch	204·2040	3318·3151
⅛	188·8887	2839·2332	⅛	204·5917	3331·0900
¼ ...	189·2814	2851·0510	¼ ...	204·9894	3343·8875
⅜	189·6741	2862·8934	⅜	205·3821	3356·7137
½ ...	190·0668	2874·7603	½ ...	205·7748	3369·5623
⅝	190·4595	2886·6517	⅝	206·1675	3382·4355
¾ ...	190·8522	2898·5677	¾ ...	206·5602	3395·3332
⅞	191·2419	2910·5083	⅞	206·9529	3408·2555
61 inch	191·6376	2922·4734	66 inch	207·3456	3421·2024
⅛	192·0303	2934·4630	⅛	207·7383	3434·1737
¼ ...	192·4230	2946·4771	¼ ...	208·1310	3447·1676
⅜	192·8157	2958·5159	⅜	208·5237	3468·1901
½ ...	193·2084	2970·5791	½ ...	208·9164	3473·2351
⅝	193·6011	2982·6669	⅝	209·3091	3486·3047
¾ ...	193·9931	2994·7792	¾ ...	209·7018	3499·3987
⅞	194·3865	3006·9161	⅞	210·0945	3512·5174
62 inch	194·7792	3019·0776	67 inch	210·4872	3525·6606
⅛	195·1719	3031·2625	⅛	210·8799	3538·8283
¼ ...	195·5646	3043·4740	¼ ...	211·2726	3552·0185
⅜	195·9573	3055·7091	⅜	211·6653	3565·2374
½ ...	196·3500	3067·9687	½ ...	212·0580	3578·4787
⅝	196·7427	3080·2529	⅝	212·4507	3591·7446
¾ ...	197·1354	3092·5615	¾ ...	212·8434	3605·0550
⅞	197·5281	3104·8948	⅞	213·2361	[illegible]

Diam.	Circum.	Area.	Diam.	Circum.	Area.
68 inch	213·6288	3631·6896	73 inch	229·3368	4185·3966
⅛......	214·0215	3645·0536	⅛......	229·7295	4199·7424
¼...	214·4142	3658·4402	¼...	230·1222	4214·1107
⅜......	214·8069	3671·8554	⅜......	230·5149	4228·5077
½...	215·1996	3685·2931	½..	230·9076	4242·9271
⅝......	215·5923	3698·7554	⅝......	231·3003	4257·3711
¾...	215·9850	3712·2421	¾...	231·6930	4271·8396
⅞......	216·3777	3725·7535	⅞......	232·0857	4286·3327
69 inch	216·7704	3739·2894	74 inch	232·4784	4300·8504
⅛......	217·1631	3752·8498	⅛......	232·8711	4315·3926
¼...	217·5558	3766·4327	¼...	233·2638	4329·9572
⅜......	217·9485	3780·0443	⅜......	233·6565	4344·5505
½...	218·3412	3793·6783	½...	234·0492	4359·1663
⅝......	218·7339	3807·3369	⅝......	234·4419	4373·8067
¾...	219·1266	3821·0200	¾...	234·8346	4388·4715
⅞......	219·5193	3834·7277	⅞......	235·2273	4403·1610
70 inch	219·9120	3848·4600	75 inch	235·6200	4417·8750
⅛......	220·3047	3862·2167	⅛......	236·0127	4432·6135
¼...	220·6974	3875·9960	¼...	236·4054	4447·3745
⅜......	221·0901	3889·8039	⅜......	236·7981	4462·1642
½...	221·4828	3903·6343	½...	237·1908	4476·9763
⅝......	221·8755	3917·4893	⅝......	237·5835	4491·8130
¾...	222·2682	3931·3687	¾...	237·9762	4506·6742
⅞......	222·6609	3945·2728	⅞......	238·3689	4521·5600
71 inch	223·0536	3959·2014	76 inch	238·7616	4536·4704
⅛......	223·4463	3973·1545	⅛......	239·1543	4551·4023
¼...	223·8390	3987·1301	¼...	239·5470	4566·3626
⅜......	224·2317	4001·1344	⅜......	239·9397	4581·3486
½...	224·6244	4015·1611	½...	240·3324	4596·3571
⅝......	225·0171	4029·2124	⅝......	240·7251	4611·3902
¾...	225·4098	4043·2882	¾...	241·1178	4626·4477
⅞......	225·8025	4057·3886	⅞......	241·5105	4641·5299
72 inch	226·1952	4071·5136	77 inch	241·9032	4656·6366
⅛......	226·5879	4085·6631	⅛......	242·2959	4671·7678
¼...	226·9806	4099·8350	¼...	242·6886	4686·9215
⅜......	227·3733	4114·0356	⅜......	243·0813	4702·1039
½...	227·7660	4128·2587	½...	243·4740	4717·3087
⅝......	228·1587	4142·5064	⅝......	243·8667	4732·5381
¾...	228·5514	4156·7785	¾...	244·2594	4747·7920
⅞......	228·9441	4171·0753	⅞......	244·6521	4763·0705

Diam.	Circum.	Area.	Diam.	Circum.	Area.
78 inch	245·0448	4778·3736	83 inch	260·7528	5410·6206
$\frac{1}{8}$	245·4375	4793·7012	$\frac{1}{8}$	261·1455	5426·9299
$\frac{1}{4}$ ...	245·8302	4809·0512	$\frac{1}{4}$ ...	261·5382	5443·2617
$\frac{3}{8}$	246·2229	4824·4299	$\frac{3}{8}$	261·9309	5459·6222
$\frac{1}{2}$ ...	246·6156	4839·8311	$\frac{1}{2}$ ...	262·3236	5476·0051
$\frac{5}{8}$	247·0083	4855·2568	$\frac{5}{8}$	262·7163	5492·4118
$\frac{3}{4}$ ...	247·4010	4870·7071	$\frac{3}{4}$ ..	263·1090	5508·8446
$\frac{7}{8}$	247·7937	4886·1820	$\frac{7}{8}$	263·5017	5525·3012
79 inch	248·1864	4901·6814	84 inch	263·8944	5541·7824
$\frac{1}{8}$	248·5791	4917·2053	$\frac{1}{8}$	264·2871	5558·2881
$\frac{1}{4}$ ...	248·9718	4932·7517	$\frac{1}{4}$ ...	264·6798	5574·8162
$\frac{3}{8}$	249·3645	4948·3268	$\frac{3}{8}$	265·0725	5591·3730
$\frac{1}{2}$ ...	249·7572	4963·9243	$\frac{1}{2}$ ...	265·4652	5607·9523
$\frac{5}{8}$	250·1499	4979·5456	$\frac{5}{8}$	265·8579	5624·5554
$\frac{3}{4}$ ...	250·5426	4995·1930	$\frac{3}{4}$ ...	266·2506	5641·1845
$\frac{7}{8}$	250·9353	5010·8642	$\frac{7}{8}$	266·6433	5657·8357
80 inch	251·3280	5026·5600	85 inch	267·0360	5674·5150
$\frac{1}{8}$	251·7207	5042·2803	$\frac{1}{8}$	267·4287	5691·2170
$\frac{1}{4}$ ...	252·1134	5058·0230	$\frac{1}{4}$ ...	267·8214	5707·9415
$\frac{3}{8}$	252·5061	5073·7944	$\frac{3}{8}$	268·2141	5724·6947
$\frac{1}{2}$ ...	252·8988	5089·5883	$\frac{1}{2}$ ...	268·6068	5741·4703
$\frac{5}{8}$	253·2915	5106·4060	$\frac{5}{8}$	268·9997	5758·2697
$\frac{3}{4}$ ...	253·6842	5121·2497	$\frac{3}{4}$ ...	269·3922	5775·0952
$\frac{7}{8}$	254·0769	5137·1173	$\frac{7}{8}$	269·7849	5791·9445
81 inch	254·4696	5153·0094	86 inch	270·1776	5808·8184
$\frac{1}{8}$	254·8623	5168·9260	$\frac{1}{8}$	270·5703	5825·7168
$\frac{1}{4}$ ..	255·2550	5184·8651	$\frac{1}{4}$ ...	270·9630	5842·6376
$\frac{3}{8}$	255·6477	5200·8329	$\frac{3}{8}$	271·3557	5859·5871
$\frac{1}{2}$ ...	256·0404	5216·8231	$\frac{1}{2}$ ...	271·7484	5876·5591
$\frac{5}{8}$	256·4331	5232·8371	$\frac{5}{8}$	272·1411	5893·5549
$\frac{3}{4}$ ...	256·8258	5248·8772	$\frac{3}{4}$ ...	272·5338	5910·5767
$\frac{7}{8}$	257·2105	5264·9411	$\frac{7}{8}$	272·9265	5927·6224
82 inch	257·6112	5281·0296	87 inch	273·3192	5944 6926
$\frac{1}{8}$	258·0039	5297·1426	$\frac{1}{8}$	273·7119	5961·7873
$\frac{1}{4}$ ...	258·3966	5313·2780	$\frac{1}{4}$ ...	274·1046	5978 9045
$\frac{3}{8}$	258·7893	5329·4421	$\frac{3}{8}$	274·4973	5996·0504
$\frac{1}{2}$ ...	259·1820	5345·6287	$\frac{1}{2}$ ...	274·8900	6013·2187
$\frac{5}{8}$	259·5747	5361·8391	$\frac{5}{8}$	275·2827	6030·4108
$\frac{3}{4}$ ...	259·9674	5378·0755	$\frac{3}{4}$ ...	275·6754	6047·6290
$\frac{7}{8}$	260·3601	5394·3358	$\frac{7}{8}$	276·0681	6064·8710

Diam.	Circum.	Area.	Diam.	Circum.	Area.
88 inch	276·4608	6082·1376	93 inch	292·1688	6792·9248
⅛	276·8535	6099·4287	⅛	292·5615	6811·1974
¼ ...	277·2462	6116·7422	¼ ...	292·9542	6829·4927
⅜	277·6389	6134·0841	⅜	293·3469	6847·8167
½ ...	278·0316	6151·4491	½ ...	293·7396	6866·1631
⅝	278·4243	6169·8376	⅝	294·1323	6884·5338
¾ ...	278·8170	6186·2591	¾ ...	294·5350	6902·9296
⅞	279·2097	6203·6905	⅞	294·9177	6921·3497
89 inch	279·6024	6221·1534	94 inch	295·3104	6939·7946
⅛	279·9951	6238·6408	⅛	295·7031	6958·2636
¼ ...	280·3878	6256·1507	¼ ...	296·0958	6976·7552
⅜	280·7805	6273·6893	⅜	296·4885	6995·2755
½ ...	281·1732	6291·2503	½ ...	296·8812	7013·8183
⅝	281·5659	6308·8351	⅝	297·2739	7032·3853
¾ ...	281·9586	6326·4460	¾ ...	297·6666	7050·9775
⅞	282·3513	6344·0807	⅞	298·0593	7069·5940
90 inch	282·7440	6361·7400	95 inch	298·4520	7088·2352
⅛	283·1367	6379·4238	⅛	298·8447	7106·9005
¼ ...	283·5294	6397·1300	¼ ...	299·2374	7125·5885
⅜	283·9221	6414·8649	⅜	299·6301	7144·3052
½ ...	284·3148	6432·6223	½ ...	300·0228	7163·0443
⅝	284·7075	6450·4039	⅝	300·4155	7181·8077
¾ ...	285·1002	6468·2107	¾ ...	300·8082	7200·5962
⅞	285·4929	6486·0418	⅞	301·2009	7219·4090
91 inch	285·8856	6503·8974	96 inch	301·5936	7238·2466
⅛	286·2783	6521·7772	⅛	301·9863	7257·1083
¼ ...	286·6710	6539·6801	¼ ...	302·3790	7275·9926
⅜	287·0637	6557·6114	⅜	302·7717	7294·9056
½ ...	287·4564	6573·5651	½ ...	303·1644	7313·8411
⅝	287·8491	6593·5431	⅝	303·5571	7332·8008
¾ ...	288·2418	6611·5462	¾ ...	303·9498	7351·7857
⅞	288·6345	6629·5736	⅞	304·3425	7370·7949
92 inch	289·0272	6647·6258	97 inch	304·7352	7389·8288
⅛	289·4199	6665·7021	⅛	305·1279	7408·8868
¼ ...	289·8125	6683·8010	¼ ...	305·5206	7427·9675
⅜	290·2053	6701·9286	⅜	305·9133	7447·0769
½ ...	290·5980	6720·0787	½ ...	306·3060	7466·2087
⅝	290·9907	6738·2530	⅝	306·6987	7485·3648
¾ ...	291·3834	6756·4525	¾ ...	307·0914	7504·5460
⅞	291·7661	6774·6763	⅞	307·4841	7523·7515

Diam.	Circum.	Area.	Diam.	Circum.	Area.
98 inch	307·8768	7542·9818	$\frac{1}{8}$	311·4111	7717·1563
$\frac{1}{8}$	308·2695	7562·2362	$\frac{1}{4}$ ...	311·8038	7736·6297
$\frac{1}{4}$ ...	308·6622	7581·5132	$\frac{3}{8}$	312·1965	7756·1318
$\frac{3}{8}$	309·0549	7600·8189	$\frac{1}{2}$ ..	312·5892	7775·6563
$\frac{1}{2}$ ...	309·4476	7620·1471	$\frac{5}{8}$	312·9819	7795·2051
$\frac{5}{8}$	309·8403	7639·4995	$\frac{3}{4}$ ...	313·3746	7814·7790
$\frac{3}{4}$ ..	310·2330	7658·8771	$\frac{7}{8}$	313·7673	7834·3772
$\frac{7}{8}$	310·6257	7678·2790			
			100 in.	314·1600	7854·0000
99 inch	311·0184	7697·7056			

TABLE II.

PROPERTIES OF SATURATED STEAM, AT TEMPERATURES FROM 32° TO 213·07° FAHRENHEIT.

Heat, in Degrees Fahrenheit.			Elastic Force.			Volume, that of an equal weight of water at its greatest density being 1.	Weight of One Cubic Foot, in Decimals of a Pound.	Specific Gravity, the Atmosphere at 32° being 1.
Sensible Heat.	Latent Heat.	Total Heat.	In Pounds on the Sq. Inch	In Inches of Mercury at 32°.	In Millimetres of Mercury at 32°.			
32	1091·70	1123·70	·089	·1811	4·6	205500	·0003	·0037
33	1091·	1124·	·092	·1884	4·786	198200	·0003	·0039
34	1090·31	1124·31	·096	·1960	4·979	191200	·0003	·0040
35	1089·61	1124·61	·100	·2039	5·179	184400	·0003	·0042
36	1088·92	1124·92	·104	·2121	5·386	177800	·0003	·0044
37	1088·22	1125·22	·108	·2205	5·6	171400	·0003	·0045
38	1087·53	1125·53	·112	·2292	5·822	165200	·0003	·0047
39	1086·83	1125·83	·117	·2382	6·051	159200	·0004	·0049
40	1086·14	1126·14	·122	·2476	6·288	153400	·0004	·0051
41	1085·44	1126·44	·127	·2573	6·534	147800	·0004	·0052
42	1084·75	1126·75	·132	·2673	6·789	142500	·0004	·0054
43	1084·05	1127·05	·137	·2777	7·053	137400	·0004	·0056
44	1083·36	1127·36	·142	·2884	7·325	132500	·0004	·0058
45	1082·66	1127·66	·147	·2994	7·606	127800	·0005	·0060
46	1081·97	1127·97	·152	·3109	7·897	123300	·0005	·0063
47	1081·27	1128·27	·158	·3228	8·199	119100	·0005	·0065
48	1080·58	1128·58	·164	·3351	8·511	115000	·0005	·0068
49	1079·88	1128·88	·170	·3478	8·833	111000	·0005	·0070
50	1079·19	1129·19	·176	·3608	9·165	107200	·0006	·0072

Heat, in Degrees Fahrenheit.			Elastic Force.			Volume, that of an equal weight of water at its greatest density being 1.	Weight of One Cubic Foot, in Decimals of a Pound.	Specific Gravity, the Atmosphere at 32° being 1.
Sensible Heat.	Latent Heat.	Total Heat.	In Pounds on the Sq. Inch.	In Inches of Mercury at 32°.	In Millimetres of Mercury at 32°.			
51	1078·49	1129·49	·183	·3743	9·508	103600	·0006	·0074
52	1077·80	1129·80	·190	·3883	9·864	100100	·0006	·0077
53	1077·10	1130·10	·197	·4028	10·232	96800	·0006	·0079
54	1076·41	1130·41	·205	·4177	10·611	93600	·0006	·0082
55	1075·71	1130·71	·212	·4332	11·002	90500	·0007	·0085
56	1075·02	1131·02	·220	·4492	11·406	87500	·0007	·0088
57	1074·32	1131·32	·228	·4656	11·823	84600	·0007	·0091
58	1073·63	1131·63	·236	·4825	12·254	81800	·0007	·0094
59	1072·93	1131·93	·245	·5000	12·699	79100	·0008	·0098
60	1072·24	1132·24	·254	·5180	13·159	76470	·0008	·0101
61	1071·54	1132·54	·263	·5367	13·633	73930	·0008	·0105
62	1070·85	1132·85	·273	·5560	14·122	71470	·0008	·0109
63	1070·15	1133·15	·282	·5758	14·626	69090	·0009	·0113
64	1069·46	1133·46	·292	·5962	15·146	66790	·0009	·0117
65	1068·76	1133·76	·302	·6173	15·682	64590	·0009	·0121
66	1068·07	1134·07	·313	·6391	16·234	62470	·0010	·0125
67	1067·37	1134·37	·324	·6615	16·803	60470	·0010	·0129
68	1066·68	1134·68	·335	·6846	17·390	58540	·0010	·0133
69	1065·98	1134·98	·347	·7084	17·996	56680	·0011	·0137
70	1065·29	1135·29	·359	·7330	18·621	54880	·0011	·0141
71	1064·59	1135·59	·372	·7583	19·265	53140	·0011	·0146
72	1063·90	1135·90	·385	·7844	19·928	51450	·0012	·0151
73	1063·20	1136·20	·398	·8114	20·611	49820	·0012	·0156
74	1062·51	1136·51	·411	·8391	21·314	48260	·0013	·0161
75	1061·81	1136·81	·425	·8676	22·037	46780	·0013	·0166

76	1061·12	1137·12	·440	·8969	22·782	45340	·0014	·0171
77	1060·42	1137·42	·455	·9171	23·550	43940	·0014	·0176
78	1059·73	1137·73	·470	·9583	24·341	42570	·0015	·0181
79	1059·03	1138·03	·486	·9905	25·155	41260	·0015	·0187
80	1058·34	1138·34	·502	1·023	25·993	40000	·0016	·0193
81	1057·64	1138·64	·518	1·056	26·855	38800	·0016	·0199
82	1056·95	1138·95	·535	1·091	27·741	37640	·0017	·0206
83	1056·25	1139·25	·553	1·127	28·653	36510	·0018	·0213
84	1055·56	1139·56	·571	1·163	29·591	35420	·0018	·0220
85	1054·86	1139·86	·590	1·201	30·556	34370	·0019	·0227
86	1054·17	1140·17	·609	1·240	31·548	33350	·0019	·0234
87	1053·47	1140·47	·629	1·281	32·568	32360	·0020	·0241
88	1052·78	1140·78	·650	1·323	33·616	31400	·0020	·0249
89	1052·08	1141·08	·671	1·366	34·694	30470	·0021	·0257
90	1051·39	1141·39	·692	1·410	35·802	29570	·0021	·0265
91	1050·69	1141·69	·714	1·454	36·942	28700	·0022	·0273
92	1050·00	1142·00	·737	1·500	38·113	27860	·0022	·0281
93	1049·30	1142·30	·760	1·548	30·317	27060	·0023	·0289
94	1048·61	1142·61	·784	1·597	40·555	26290	·0023	·0297
95	1047·91	1142·91	·808	1·647	41·827	25540	·0024	·0305
96	1047·22	1143·22	·833	1·698	43·134	24810	·0024	·0314
97	1046·52	1143·52	·859	1·751	44·476	24100	·0025	·0323
98	1045·83	1143·83	·886	1·805	45·854	23410	·0025	·0332
99	1045·13	1144·13	·913	1·861	47·268	22740	·0026	·0341
100	1044·44	1144·44	·942	1·918	48·719	22100	·0027	·0350
101	1043·74	1144·74	·971	1·977	50·207	21480	·0028	·0360
102	1043·05	1145·05	1·000	2·037	51·734	20890	·0029	·0370
103	1042·35	1145·35	1·030	2·099	53·3	20310	·0030	·0381
	1041·66	1145·66	1·061	2·163	54·906	19750	·0031	·0392
	1040·96	1145·96	1·093	2·227	56·554	19210	·0032	·0403
	1040·27	1146·27	1·125	2·293	58·245	18690	·0033	·0415

Heat, in Degrees Fahrenheit.			Elastic Force.			Volume, that of an equal weight of water at its greatest density being 1.	Weight of One Cubic Foot, in Decimals of a Pound.	Specific Gravity, the Atmosphere at 32° being 1.
Sensible Heat.	Latent Heat.	Total Heat.	In Pounds on the Sq. Inch	In Inches of Mercury at 32°.	In Millimetres of Mercury at 32°.			
107	1039·57	1146·57	1·158	2·361	59·981	18180	·0034	·0427
108	1038·88	1146·88	1·192	2·431	61·763	17690	·0035	·0439
109	1038·18	1147·18	1·228	2·503	63·592	17210	·0036	·0451
110	1037·49	1147·49	1·264	2·577	65·469	16740	·0037	·0463
111	1036·79	1147·79	1·301	2·653	67·394	16280	·0038	·0475
112	1036·10	1148·10	1·340	2·731	69·368	15840	·0039	·0488
113	1035·40	1148·40	1·380	2·810	71·391	15420	·0040	·0501
114	1034·71	1148·71	1·420	2·892	73·464	15010	·0041	·0514
115	1034·01	1149·01	1·461	2·976	75·588	14620	·0042	·0528
116	1033·32	1149·32	1·503	3·061	77·764	14240	·0043	·0542
117	1032·62	1149·62	1·546	3·149	79·993	13870	·0044	·0557
118	1031·93	1149·93	1·590	3·239	82·276	13510	·0045	·0572
119	1031·23	1150·23	1·635	3·331	84·615	13160	·0046	·0587
120	1030·54	1150·54	1·681	3·425	87·012	12820	·0048	·0603
121	1029·84	1150·84	1·729	3·522	89·467	12490	·0049	·0619
122	1029·15	1151·15	1·778	3·621	91·982	12170	·0051	·0635
123	1028·45	1151·45	1·827	3·723	94·557	11860	·0052	·0652
124	1027·76	1151·76	1·878	3·826	97·194	11560	·0054	·0669
125	1027·06	1152·06	1·930	3·933	99·893	11270	·0055	·0686
126	1026·37	1152·37	1·983	4·042	102·656	10980	·0057	·9704
127	1025·67	1152·67	2·038	4·153	105·484	10700	·0058	·0722
128	1024·98	1152·98	2·094	4·267	108·379	10430	·0060	·0741
129	1024·28	1153·28	2·152	4·384	111·342	10170	·0061	·0760
130	1023·59	1153·59	2·211	4·503	114·375	9920	·0063	·0780
131	1022·89	1153·89	2·271	4·625	117·478	9670	·0064	0·800

132	1022·20	1154·20	2·333	4·750	120·653	9430	·0066	·0820
133	1021·50	1154·50	2·396	4·878	123·901	9200	·0067	·0841
134	1020·81	1154·81	2·459	5·009	127·223	8980	·0069	·0862
135	1020·11	1155·11	2·524	5·143	130·621	8760	·0070	·0883
136	1019·42	1155·42	2·592	5·280	134·096	8550	·0072	·0905
137	1018·72	1155·72	2·661	5·420	137·650	8340	·0074	·0927
138	1018·03	1156·03	2·730	5·563	141·283	8140	·0076	·0950
139	1017·33	1156·33	2·801	5·709	144·996	7950	·0078	·0973
140	1016·64	1156·64	2·874	5·858	148·791	7760	·0080	·0997
141	1015·94	1156·94	2·950	6·011	152·670	7580	·0082	·1021
142	1015·25	1157·25	3·027	6·167	156·635	7400	·0084	·1046
143	1014·55	1157·55	3·106	6·327	160·688	7224	·0086	·1071
144	1013·86	1157·86	3·186	6·490	164·831	7053	·0088	·1097
145	1013·16	1158·16	3·268	6·657	169·065	6887	·0090	·1123
146	1012·47	1158·47	3·351	6·827	173·392	6725	·0092	·1150
147	1011·77	1158·77	3·436	7·001	177·814	6568	·0094	·1177
148	1011·08	1159·08	3·523	7·179	182·331	6415	·0097	·1205
149	1010·38	1159·38	3·612	7·361	186·954	6266	·0099	·1233
150	1009·69	1159·69	3·704	7·547	191·658	6122	·0102	·1262
151	1008·99	1159·99	3·798	7·736	196·471	5982	·0104	·1292
152	1008·30	1160·30	3·893	7·929	201·386	5846	·0107	·1322
153	1007·60	1160·60	3·989	8·127	206·404	5713	·0109	·1353
154	1006·91	1160·91	4·087	8·329	211·527	5584	·0112	·1385
155	1006·21	1161·21	4·188	8·535	216·756	5458	·0114	·1417
156	1005·52	1161·52	4·292	8·745	222·092	5335	·0117	·1450
157	1004·82	1161·82	4·397	8·958	228·537	5215	·0119	·1483
158	1004·13	1162·13	4·504	9·178	233·093	5098	·0122	·1517
159	1003·43	1162·43	4·614	9·401	238·763	4984	·0125	·1551
160	1002·74	1162·74	4·726	9·629	244·551	4874	·0128	·1586
161	1002·04	1163·04	4·840	9·861	250·459	4768	·0131	·1622
162	1001·35	1163·35	4·956	10·098	256·488	4665	·0134	·1658

Heat, in Degrees Fahrenheit.			Elastic Force.			Volume, that of an equal weight of water at its greatest density being 1.	Weight of One Cubic Foot, in Decimals of a Pound.	Specific Gravity, the Atmosphere at 32° being 1.
Sensible Heat.	Latent Heat.	Total Heat.	In Pounds on the Sq. Inch.	In Inches of Mercury at 32°.	In Millimetres of Mercury at 32°.			
163	1000·65	1163·65	5·075	10·340	262·641	4564	·0137	·1695
164	999·96	1163·96	5·196	10·588	268·919	4465	·0140	·1732
165	999·26	1164·26	5·320	10·840	275·323	4368	·0143	·1770
166	998·57	1164·57	5·447	11·097	281·855	4274	·0146	·1809
167	997·87	1264·87	5·576	11·359	288·517	4182	·0149	·1849
168	997·18	1165·18	5·707	11·627	295·311	4092	·0152	·1889
169	996·48	1165·48	5·841	11·900	302·238	4005	·0155	·1930
170	995·79	1165·79	5·977	12·178	309·300	3920	·0159	·1972
171	995·09	1166·09	6·116	12·461	316·499	3837	·0162	·2015
172	994·40	1166·40	6·258	12·750	323·838	3756	·0166	·2058
173	993·70	1166·70	6·403	13·045	331·319	3677	·0169	·2102
174	993·01	1167·01	6·550	13·345	338·945	3600	·0173	·2147
175	992·31	1167·31	6·699	13·651	346·719	3525	·0176	·2193
176	991·62	1167·62	6·852	13·963	354·643	3452	·0180	·2240
177	990·92	1167·92	7·008	14·281	362·719	3381	·0184	·2287
178	990·23	1168·23	7·167	14·605	370·949	3311	·0188	·2335
179	989·53	1168·53	7·330	14·935	379·335	3243	·0192	·2384
180	988·84	1168·84	7·495	15·271	387·879	3176	·0196	·2434
181	988·14	1169·14	7·663	15·614	396·583	3111	·0200	·2485
182	987·45	1169·45	7·835	15·963	405·449	3047	·0204	·2537
183	986·75	1169·75	8·010	16·318	414·479	2985	·0208	·2590
184	986·06	1170·06	8·188	16·680	423·676	2925	·0213	·2643
185	985·36	1170·36	8·369	17·049	433·041	2866	·0217	·2697
186	984·67	1170·67	8·552	17·425	442·578	2809	·0222	·2752
187	983·97	1170·97	8·739	17·807	452·291	2758	·0226	·2808

188	983·28	1171·28	8·930	18·196	462·188	2698	·0231	·2865
189	982·58	1171·58	9·125	18·593	472·257	2645	·0235	·2923
190	981·89	1171·89	9·323	18·997	482·516	2593	·0240	·2982
191	981·19	1172·19	9·525	19·408	492·963	2542	·0245	·3042
192	980·50	1172·50	9·731	19·827	503·600	2492	·0250	·3103
193	979·80	1172·80	9·940	20·253	514·430	2443	·0255	·3165
194	979·11	1173·11	10·153	20·687	525·455	2395	·0260	·3228
195	978·41	1173·41	10·370	21·129	536·677	2348	·0265	·3292
196	977·72	1173·72	10·590	21·579	548·098	2302	·0271	·3358
197	977·02	1174·02	10·814	22·036	559·720	2257	·0276	·3425
198	976·33	1174·33	11·043	22·502	571·545	2213	·0282	·3493
199	975·63	1174·63	11·276	22·976	583·575	2171	·0287	·3561
200	974·94	1174·94	11·513	23·458	595·812	2130	·0293	·3630
201	974·24	1175·24	11·753	23·948	608·258	2090	·0298	·3700
202	973·55	1175·55	11·997	24·446	620·915	2051	·0304	·3771
203	972·85	1175·85	12·246	24·953	633·785	2013	·0310	·3843
204	972·16	1176·16	12·499	25·468	646·873	1975	·0316	·3916
205	971·46	1176·46	12·757	25·992	660·184	1938	·0322	·3990
206	970·77	1176·77	13·019	26·525	673·723	1902	·0328	·4065
207	970·07	1177·07	13·286	27·067	687·495	1867	·0334	·4141
208	969·38	1177·38	13·558	27·619	701·505	1833	·0340	·4219
209	968·68	1177·68	13·833	28·180	715·757	1799	·0346	·4298
210	967·99	1177·99	14·112	28·751	730·254	1766	·0353	·4378
211	967·29	1178·29	14·396	29·332	745·000	1734	·0359	·4460
212	966·60	1178·60	14·685	29·9218	760·000	1702	·0366	·4543
213	965·90	1178·90	14·980	30·522	775·234	1671	·0373	·4627
213·07	965·85	1178·92	15·000	30·562	776·273	1669	·0373	·4633

TABLE III.

PROPERTIES OF SATURATED STEAM, AT PRESSURES FROM ONE POUND TO TWO HUNDRED POUNDS ON THE SQUARE INCH.

Elastic Force.		Heat, in Degrees Fahrenheit.					Volume, that of an equal weight of water at its greatest density being 1.	Weight of One Cubic Foot, in Decimals of a Pound.	Specific Gravity, the Atmosphere at 32° being 1.
In Pounds on the Sq. Inch	In Inches of Mercury at 32°.	Sensible Heat.		Latent Heat.	Total Heat.				
			Difference.			Difference.			
1	2·0375	102·		1043·05	1145·05		20890	·0029	·037
2	4·075	126·31	24·31	1026·15	1152·46	7·41	10890	·0057	·071
3	6·1125	141·65	15·34	1015·49	1157·14	4·68	7460	·0084	·104
4	8·15	153·11	11·46	1007·53	1160·64	3·50	5700	·0110	·136
5	10·1875	162·37	9·26	1001·09	1163·46	2·82	4627	·0135	·167
6	12·225	170·17	7·80	995·67	1165·84	2·38	3906	·0160	·198
7	14·2625	176·95	6·78	990·96	1167·91	2·07	3385	·0185	·228
8	16·3	182·95	6·00	986·79	1169·74	1·83	2988	·0209	·258
9	18·3375	188·36	5·41	983·03	1171·39	1·65	2678	·0233	·288
10	20·375	193·29	4·93	979·60	1172·89	1·50	2429	·0257	·318
11	22·4125	197·82	4·53	976·45	1174·27	1·38	2224	·0281	·348
12	24·45	201·01	4·19	973·54	1175·55	1·28	2052	·0304	·377
13	26·4875	205·92	3·91	970·83	1176·75	1·20	1905	·0327	·406
14	28·525	209·6	3·68	968·27	1177·87	1·12	1779	·0350	·435
15	30·5625	213·07	3·47	965·85	1178·92	1·05	1669	·0373	·463
16	32·6	216·3	3·23	963·6	1179·9	1·02	1572	·0396	·492
17	34·6375	219·4	3·1	961·5	1180·9	1·0	1486	·0419	·520
18	36·675	222·4	3·0	959·4	1181·8	·9	1410	·0442	·548
19	38·7125	225·2	2·8	957·5	1182·7	·9	1342	·0465	·576
20	40·75	228·	2·8	955·5	1188·5	·8	1280	·0487	·604

21	**42·7175**	**230·6**	**2·3**	**953·7**	**1184·3**	**·3**	**1224**	·0510	**·632**
22	**44·825**	**233·1**	**2·5**	**951·9**	**1185·0**	**·7**	**1172**	·0532	**·660**
23	**46·8625**	**235·5**	**2·4**	**950·2**	**1185·7**	**·7**	**1125**	·0554	**·688**
24	**48·9**	**237·9**	**2·4**	**948·6**	1186·5	**·8**	**1082**	·0576	**·715**
25	50·9375	240·2	2·3	947·0	1187·2	·7	1042	·0593	·742
26	52·975	242·3	2·1	945·6	1187·9	·7	1005	·0620	·769
27	55·0125	244·4	2·1	944·1	1188·5	·6	971	·0642	·796
28	57·95	246·4	2·	942·7	1189·1	·6	939	·0664	·823
29	59·0875	248·4	2·	941·3	1189·7	·5	909	·0686	·850
30	61·125	250·4	2·	939·9	1190·3	·6	881	·0707	·877
31	63·1625	252·3	1·9	938·5	1190·8	·5	855	·0726	·904
32	65·2	254·1	1·8	937·3	1191·4	·6	830	·0751	·931
33	67·2357	255·9	1·8	936·1	1192·	·6	807	·0772	·958
34	69·275	257·6	1·7	934·9	1192·5	·5	785	·0794	·985
35	71·3125	259·3	1·7	933·7	1193·	·5	764	·0815	1·012
36	73·35	260·9	1·6	932·6	1193·5	·5	745	·0837	1·038
37	75·3875	262·6	1·7	931·4	1194·	·5	727	·0858	1·064
38	77·425	264·2	1·6	930·3	1194·5	·5	709	·0879	1·090
39	79·4625	265·8	1·6	929·2	1195·	·5	692	·0900	1·116
40	81·5	267·3	1·5	928·1	1195·4	·4	676	·0921	1·142
41	83·5357	268·7	1·4	927·2	1195·9	·5	661	·0942	1·168
42	85·575	270·2	1·5	926·1	1196·3	·4	647	·0963	1·194
43	87·6125	271·6	1·4	925·2	1196·8	·5	634	·0983	1·220
44	89·65	273·	1·4	924·2	1197·2	·4	621	·1004	1·246
45	91·6875	274·4	1·4	923·2	1197·6	·4	608	·1025	1·272
46	93·725	275·8	1·4	922·2	1198·	·4	596	·1046	1·298
47	25·7625	277·1	1·3	921·3	1198·4	·4	584	·1067	1·324
48	97·8	278·4	1·3	920·4	1198·8	·4	573	·1087	1·350
49	99·8375	279·7	1·3	919·5	1199·2	·4	562	·1108	1·376

Elastic Force. In Pounds on the Sq. Inch	Elastic Force. In Inches of Mercury at 32°.	Heat, in Degrees Fahrenheit. Sensible Heat.	Sensible Heat. Difference.	Latent Heat.	Total Heat.	Total Heat. Difference.	Volume, that of an equal weight of water at its greatest density being 1.	Weight of One Cubic Foot, in Decimals of a Pound.	Specific Gravity, the Atmosphere at 32° being 1.
50	101·875	281·	1·3	918·6	1199·6	·4	552	·1129	1·402
51	103·9125	282·3	1·3	917·7	1200·	·4	542	·1150	1·428
52	105·95	283·5	1·2	916·9	1200·4	·4	532	·1171	1·454
53	107·9875	284·7	1·2	916·1	1200·8	·4	523	·1192	1·479
54	110·025	285·9	1·2	915·2	1201·1	·3	514	·1212	1·504
55	112·0625	287·1	1·2	914·4	1201·5	·4	506	·1232	1·529
56	114·1	288·2	1·1	913·6	1201·8	·3	498	·1252	1·554
57	116·1375	289·3	1·1	912·9	1202·2	·4	490	·1272	1·579
58	118·175	290·4	1·1	912·1	1202·5	·3	482	·1293	1·604
59	120·2125	291·6	1·2	911·3	1202·9	·4	474	·1314	1·629
60	122·25	292·7	1·1	910·5	1203·2	·3	467	·1335	1·654
61	124·2875	293·8	1·1	909·8	1203·6	·4	460	·1356	1·679
62	126·325	294·8	1·	909·1	1203·9	·3	453	·1376	1·704
63	128·3625	295·9	1·1	908·3	1204·2	·3	447	·1396	1·729
64	130·4	296·9	1·	907·6	1204·5	·3	440	·1416	1·754
65	132·4375	298·	1·1	906·8	1204·8	·3	434	·1436	1·779
66	134·475	299·	1·	906·1	1205·1	·3	428	·1456	1·804
67	136·5125	300·	1·	905·4	1205·4	·3	422	·1476	1·829
68	138·55	300·9	·9	904·8	1205·7	·3	417	·1496	1·854
69	140·5875	301·9	1·	904·1	1206·	·3	411	·1516	1·879
70	142·625	302·9	1·	903·4	1206·3	·3	406	·1536	1·904
71	144·6625	303·9	1·	902·7	1206·6	·3	401	·1556	1·929
72	146·7	304·8	·9	902·1	1206·9	·3	396	·1576	1·954
73	148·7375	305·7	·9	901·5	1207·2	·3	391	·1596	1·979

74	150·775	306·6	·9	900·9	1207·5	·3	386	·1616	2·004
75	152·8125	307·5	·9	900·3	1207·8	·3	381	·1636	2·029
76	154·85	308·4	·9	899·6	1208·	·2	377	·1656	2·054
77	156·8875	309·3	·9	899·0	1208·3	·3	372	·1676	2·079
78	158·925	310·2	·9	898·4	1208·6	·3	368	·1696	2·103
79	160·9625	311·1	·9	897·8	1208·9	·3	364	·1716	2·127
80	163·	312·	·9	897·1	1209·1	·2	359	·1736	2·151
81	165·0375	312·8	·8	896·6	1209·4	·3	355	·1756	2·175
82	167·075	313·6	·8	896·1	1209·7	·3	351	·1776	2·199
83	169·1125	314·5	·9	895·4	1209·9	·2	348	·1795	2·223
84	171·15	315·3	·8	894·8	1210·1	·2	344	·1814	2·247
85	173·1875	316·1	·8	894·3	1210·4	·3	340	·1833	2·271
86	175·225	316·9	·8	893·8	1210·7	·3	337	·1852	2·295
87	177·2625	317·8	·9	893·1	1210·9	·2	333	·1871	2·319
88	179·3	318·6	·8	892·5	1211·1	·2	330	·1891	2·343
89	181·3375	319·4	·8	892·0	1211·4	·3	326	·1910	2·367
90	183·375	320·2	·8	891·4	1211·6	·2	323	·1930	2·391
91	185·4125	321·	·8	890·8	1211·8	·2	320	·1950	2·415
92	187·45	321·7	·7	890·3	1212·0	·2	317	·1970	2·439
93	189·4875	322·5	·8	889·8	1212·3	·3	313	·1990	2·463
94	191·525	323·3	·8	889·2	1212·5	·2	310	·2010	2·487
95	193·5625	324·1	·8	888·7	1212·8	·3	307	·2030	2·511
96	195·6	324·8	·7	888·2	1213·0	·2	305	·2050	2·535
97	197·6375	325·6	·8	887·7	1213·3	·3	302	·2070	2·559
98	199·675	326·3	·7	887·2	1213·5	·2	299	·2089	2·583
99	201·7125	327·1	·8	886·6	1213·7	·2	296	·2108	2·607
100	203·75	327·8	·7	886·1	1213·9	·2	293	·2127	2·631
	205·7875	328·5	·7	885·7	1214·2	·3	290	·2147	2·655
	207·825	329·2	·7	885·2	1214·4	·2	288	·2167	2·679

Elastic Force.		Heat, in Degrees Fahrenheit.						Volume, that of an equal weight of water at its greatest density being 1.	Weight of One Cubic Foot, in Decimals of a Pound.	Specific Gravity, the Atmosphere at 32° being 1.
In Pounds on the Sq. Inch.	In Inches of Mercury at 32°.	Sensible Heat.	Difference.	Latent Heat.	Total Heat.	Difference.				
103	209·8625	329·9	·7	884·7	1214·6	·2	285	·2186	2·703
104	211·9	330·6	·7	884·2	1214·8	·2	283	·2205	2·727
105	213·9375	331·3	·7	883·7	1215·0	·2	281	·2224	2·751
106	215·975	331·9	·6	883·3	1215·2	·2	278	·2243	2·755
107	218·0125	332·6	·7	882·8	1215·4	·2	276	·2262	2·799
108	220·05	333·3	·7	882·3	1215·6	·2	273	·2281	2·823
109	222·0875	334·	·7	881·8	1215·8	·2	271	·2300	2·847
110	224·125	334·6	·6	881·4	1216·0	·2	269	·2319	2·871
111	226·1625	335·3	·7	880·9	1216·2	·2	267	·2337	2·895
112	228·2	336·	·7	880·4	1216·4	·2	265	·2355	2·919
113	230·2375	336·7	·7	879·9	1216·6	·2	263	·2374	2·943
114	232·275	337·4	·7	879·4	1216·8	·2	261	·2392	2·967
115	234·3125	338·	·6	879·0	1217·0	·2	259	·2410	2·990
116	236·35	338·6	·6	878·6	1217·2	·2	257	·2428	3·013
117	238·3875	339·3	·7	878·1	1217·4	·2	255	·2446	3·036
118	240·425	339·9	·6	877·7	1217·6	·2	253	·2465	3·059
119	242·4625	340·5	·6	877·3	1217·8	·2	251	·2484	3·082
120	244·5	341·1	·6	876·9	1218·0	·2	249	·2503	3·105
121	246·5375	341·8	·7	876·4	1218·2	·2	247	·2522	3·130
122	248·575	342·4	·6	876·0	1218·4	·2	245	·2541	3·155
123	250·6125	343·	·6	875·6	1218·6	·2	243	·2560	3·179
124	252·65	343·6	·6	875·1	1218·7	·1	241	·2579	3·203
125	254·6875	344·2	·6	874·7	1218·9	·2	239	·2598	3·227

126	256·725	344·8	·6	874·3	1219·1	·2	238	·2617	3·251
127	258·7625	345·4	·6	873·9	1219·3	·2	236	·2636	3·275
128	260·8	346·	·6	873·4	1219·4	·1	234	·2655	3·299
129	262.8375	346·6	·6	873·0	1219·6	·2	232	·2674	3·323
130	264·875	347·2	·6	872·6	1219·8	·2	231	·2693	3·347
131	266·9125	347·8	·6	872·2	1220·	·2	229	·2712	3·371
132	268·95	348·3	·5	871·9	1220·2	·2	228	·2731	3·395
133	270·9875	348·9	·6	871·5	1220·4	·2	226	·2750	3·419
134	273·025	349·5	·6	871·1	1220·6	·2	225	·2769	3·443
135	275·0625	350·	·5	870·7	1220·7	·1	223	·2788	3·467
136	277·1	350·6	·6	870·3	1220·9	·2	222	·2807	3·490
137	279·1375	351·2	·6	869·8	1221·	·1	220	·2826	3·513
138	281·175	351·8	·6	869·4	1221·2	·2	219	·2845	3·536
139	283·2125	352·3	·5	869·1	1221·4	·2	217	·2864	3·559
140	285·25	352·9	·6	868·6	1221·5	·1	216	·2883	3·582
141	287·2875	353·4	·5	868·3	1221·7	·2	214	·2902	3·605
142	289·325	354·	·6	867·9	1221·9	·2	213	·2921	3 628
143	291·3625	354·5	·5	867·5	1222·	·1	211	·2940	3·651
144	293·4	355·	·5	867·2	1222·2	·2	210	·2959	3·674
145	295 4375	355·6	·6	866·8	1222·4	·2	209	·2978	3·697
146	297·475	356·1	·5	866·4	1222·5	·1	208	·2997	3·720
147	299·5125	356·7	·6	866·0	1222·7	·2	206	·3016	3·743
148	301.55	357·2	·5	865·7	1222·9	·2	205	·3035	3·765
149	303.5875	357·8	·6	865·2	1223·	·1	204	·3054	3·787
150	305.625	358·3	·5	864·9	1223·2	·2	203	·3073	3·809
151	307·6625	358·9	·6	864·5	1223·4	·2	201	·3092	3·832
152	309·7	359·4	·5	864·1	1223·5	·1	200	·3111	3·856
153	311·7375	359.9	·5	863·8	1223·7	·2	199	·3130	3·880
154	313·775	360·4	·5	863·4	1223·8	·1	198	·3149	3·904

Elastic Force.		Heat, in Degrees Fahrenheit.						Volume, that of an equal weight of water at its greatest density being 1.	Weight of One Cubic Foot, in Decimals of a Pound.	Specific Gravity, the Atmosphere at 32° being 1.
In Pounds on the Sq. Inch.	In Inches of Mercury at 32°.	Sensible Heat.	Difference.	Latent Heat.	Total Heat.	Difference.				
155	315·8125	360·9	·5	863·1	1224·	·2	196	·3168	3·927
156	317·85	361·4	·5	862·7	1224·1	·1	195	·3187	3·950
157	319·8875	361·9	·5	862·4	1224·3	·2	194	·3206	3·973
158	321·925	362·4	5	862·0	1224·4	·1	193	·3225	3·996
159	323·9625	362·9	·5	861·7	1224·6	·2	192	·3244	4·019
160	326·	363·4	·5	861·4	1224·8	·2	191	·3263	4·042
161	328·0375	363·9	·5	861·0	1224·9	·1	190	·3281	4·065
162	330·075	364·4	·5	860·7	1225·1	·2	189	·3300	4·088
163	332·1125	364·9	·5	860·4	1225·3	·2	188	·3317	4·111
164	334·15	365·4	·5	860·0	1225·4	·1	187	·3335	4·134
165	336·1875	365·9	·5	859·7	1225·6	·2	186	·3353	4·157
166	338·225	366·4	·5	859·3	1225·7	·1	185	·3371	4·180
167	340·2625	366·9	·5	858·9	1225·8	·1	184	·3389	4·203
168	342·3	367·4	·5	858·6	1226·	·2	183	·3407	4·225
169	344·3375	367·8	·4	858·3	1226·1	·1	182	·3425	4·248
170	346·375	368·2	·4	858·0	1226·3	·2	181	·3443	4·270
171	348·4125	368·7	·5	857·7	1226·4	·1	180	·3461	4·293
172	350·45	369·2	·5	857·4	1226·6	·2	179	·3479	4·315
173	352·4875	369·7	·5	857·1	1226·7	·1	178	·3497	4·338
174	354·525	370·1	·4	856·8	1226·9	·2	177	·3515	4·360
175	356·5625	370·6	·5	856·4	1227·	·1	176	·3533	4·383
176	358·6	371·1	·5	856·1	1227·2	·2	175	·3551	4·405
177	360·6375	371·5	·4	855·8	1227·3	·1	174	·3569	4·428
178	362·675	372·	·5	855·4	1227·4	·1	173	·3587	4·450

179	364·7125	372·5	·5	855·1	1227·6	·2	172	·3605	4·473
180	366·75	372·9	·4	854·8	1227·7	·1	172	·3623	4·495
181	368·7875	373·4	·5	854·4	1227·8	·1	171	·3641	4·517
182	370·825	373·9	·5	854·1	1228·	·2	170	·3659	4·540
183	372·8625	374·3	·4	853·8	1228·1	·1	169	·3677	4·562
184	374·9	374·8	·5	853·5	1228·3	·2	169	·3695	4·585
185	376·9375	375·3	·5	853·1	1228·4	·1	168	·3713	4·607
186	378·975	375·7	·4	852·9	1228·6	·2	167	·3731	4·630
187	381·0125	376·2	·5	852·5	1228·7	·1	166	·3749	4·652
188	383·05	376·6	·4	852·2	1228·8	·1	165	·3766	4·675
189	385·0875	377·1	·5	851·9	1229·	·2	165	·3783	4·697
190	387·125	377·5	·4	851·6	1229·1	·1	164	·3800	4·720
191	389·1625	378·	·5	851·2	1229·2	·1	163	·3818	4·742
192	391·2	378·4	·4	851·0	1229·4	·2	162	·3836	4·765
193	393·2375	378·8	·4	850·7	1229·5	·1	161	·3854	4·787
194	395·275	379·3	·5	850·3	1229·6	·1	161	·3871	4·810
195	397·3125	379·7	·4	850·1	1229·8	·2	160	·3888	4·832
196	399·35	380·1	·4	849·8	1229·9	·1	159	·3905	4·855
197	401·3875	380·6	·5	849·4	1230·	·1	159	·3922	4·877
198	403·425	381·	·4	849·1	1230·1	·1	158	·3939	4·900
199	405·4625	381·4	·4	848·8	1230·2	·1	157	·3956	4·922
200	407·5	381·7	·3	848·6	1230·3	·1	157	·3973	4·945

TABLE IV.

HYPERBOLIC LOGARITHMS.

No.	Logarithm.	No.	Logarithm.	No.	Logarithm.	No.	Logarithm.
1·02	·0198	1·72	·5423	2·42	·8837	3·12	1·1378
1·04	·0392	1·74	·5539	2·44	·8920	3·14	1·1442
1·06	·0582	1·76	·5653	2·46	·9001	3·16	1·1505
1·08	·0769	1·78	·5766	2·48	·9082	3·18	1·1569
1·10	·0953	1·80	·5878	2·50	·9163	3·20	1·1631
1·12	·1133	1·82	·5988	2·52	·9242	3·22	1·1694
1·14	·1310	1·84	·6097	2·54	·9321	3·24	1·1755
1·16	·1484	1·86	·6205	2·56	·9400	3·26	1·1817
1·18	·1655	1·88	·6312	2·58	·9478	3·28	1·1878
1·20	·1823	1·90	·6418	2·60	·9555	3·30	1·1939
1·22	·1988	1·92	·6523	2·62	·9631	3·32	1·1999
1·24	·2151	1·94	·6627	2·64	·9708	3·34	1·2059
1·26	·2311	1·96	·6729	2·66	·9783	3·36	1·2119
1·28	·2468	1·98	·6831	2·68	·9858	3·38	1·2178
1·30	·2623	2·00	·6931	2·70	·9932	3·40	1·2237
1·32	·2776	2·02	·7031	2·72	1·0006	3·42	1·2296
1·34	·2926	2·04	·7129	2·74	1·0079	3·44	1·2354
1·36	·3075	2·06	·7227	2·76	1·0152	3·46	1·2412
1·38	·3221	2·08	·7323	2·78	1·0221	3·48	1·2470
1·40	·3364	2·10	·7419	2·80	1·0296	3·50	1·2527
1·42	·3506	2·12	·7514	2·82	1·0367	3·52	1·2584
1·44	·3646	2·14	·7608	2·84	1·0438	3·54	1·2641
1·46	·3784	2·16	·7701	2·86	1·0508	3·56	1·2697
1·48	·3920	2·18	·7793	2·88	1·0578	3·58	1·2753
1·50	·4054	2·20	·7884	2·90	1·0647	3·60	1·2809
1·52	·4187	2·22	·7975	2·92	1·0716	3·62	1·2864
1·54	·4318	2·24	·8064	2·94	1·0784	3·64	1·2920
1·[illegible]	·4447	2·26	·8153	2·96	1·0852	3·66	1·2974
1·58	·4574	2·28	·8242	2·98	1 0919	3·68	1·3029
1·60	·4700	2·30	·8329	3·00	1·0986	3·70	1·3083
1·62	·4824	2·32	·8415	3·02	1·1052	3·72	1·3137
1·64	·4947	2·34	·8501	3·04	1·1118	3·74	1·3191
1·66	·5068	2·36	·8586	3·06	1·1184	3·76	1·3244
1·68	·5188	2·38	·8671	3·08	1·1249	3·78	1·3297
1·70	·5306	2·40	·8754	3·10	1·1314	3·80	1·3350

No.	Logarithm.	No.	Logarithm.	No.	Logarithm.	No.	Logarithm.
3·82	1·3402	4·62	1·5304	5·42	1·6901	6·22	1·8277
3·84	1·3454	4·64	1·5347	5·44	1·6937	6·24	1·8310
3·86	1·3506	4·66	1·5390	5·46	1·6974	6·26	1·8342
3·88	1·3558	4·68	1·5433	5·48	1·7011	6·28	1·8373
3·90	1·3609	4·70	1·5475	5·50	1·7047	6·30	1·8405
3·92	1·3661	4·72	1·5518	5·52	1·7083	6·32	1·8437
3·94	1·3712	4·74	1·5560	5·54	1·7120	6·34	1·8468
3·96	1·3762	4·76	1·5602	5·56	1·7156	6·36	1·8500
3·98	1·3813	4·78	1·5644	5·58	1·7192	6·38	1·8531
4·00	1·3863	4·80	1·5686	5·60	1·7227	6·40	1·8563
4·02	1·3913	4·82	1·5727	5·62	1·7263	6·42	1·8594
4·04	1·3962	4·84	1·5769	5·64	1·7299	6·44	1·8625
4·06	1·4012	4·86	1·5810	5·66	1·7334	6·46	1·8656
4·08	1·4061	4·88	1·5851	5·68	1·7369	6·48	1·8687
4·10	1·4110	4·90	1·5892	5·70	1·7404	6·50	1·8718
4·12	1·4158	4·92	1·5933	5·72	1·7439	6·52	1·8748
4·14	1·4207	4·94	1·5973	5·74	1·7475	6·54	1·8779
4·16	1·4255	4·96	1·6014	5·76	1·7509	6·56	1·8810
4·18	1·4303	4·98	1·6054	5·78	1·7544	6·58	1·8840
4·20	1·4351	5·00	1·6094	5·80	1·7578	6·60	1·8870
4·22	1·4398	5·02	1·6134	5·82	1·7613	6·62	1·8901
4·24	1·4445	5·04	1·6174	5·84	1·7647	6·64	1·8931
4·26	1·4492	5·06	1·6213	5·86	1·7681	6·66	1·8961
4·28	1·4539	5·08	1·6253	5·88	1·7715	6·68	1·8991
4·30	1·4586	5·10	1·6292	5·90	1·7749	6·70	1·9021
4·32	1·4632	5·12	1·6331	5·92	1·7783	6·72	1·9051
4·34	1·4678	5·14	1·6370	5·94	1·7817	6·74	1·9080
4·36	1·4724	5·16	1·6409	5·96	1·7850	6·76	1·9110
4·38	1·4770	5·18	1·6448	5·98	1·7884	6·78	1·9139
4·40	1·4816	5·20	1·6486	6·00	1·7917	6.80	1·9169
4·42	1·4861	5·22	1·6525	6·02	1·7951	6·82	1·9198
4·44	1·4906	5·24	1·6563	6·04	1·7984	6·84	1·[illegible]28
4·46	1·4951	5·26	1·6601	6·06	1·8017	6·86	1·9257
4·48	1·4996	5·28	1·6639	6·08	1·8050	6·88	1·9286
4·50	1·5040	5·30	1·6677	6·10	1·8083	6·90	1·9315
4·52	1·5085	5·32	1·6714	6·12	1·8115	6·92	1·9344
4·54	1·5129	5·34	1·6752	6·14	1·8148	6·94	1·9373
4·56	1·5173	5·36	1·6789	6·16	1·8180	6·96	1·9401
4·58	1·5217	5·38	1·6827	6·18	1·8213	6·98	1·9430
4·60	1·5260	5·40	1·6864	6·20	1·8245	7.00	1·9459

No.	Logarithm.	No.	Logarithm.	No.	Logarithm.	No.	Logarithm.
7·02	1·9487	7·82	2·0567	8·62	2·1541	9·42	2·2428
7·04	1·9516	7·84	2·0592	8·64	2·1564	9·44	2·2449
7·06	1·9544	7·86	2·0618	8·66	2·1587	9·46	2·2470
7·08	1·9572	7·88	2·0643	8·68	2·1610	9·48	2·2492
7·10	1·9600	7·90	2·0668	8·70	2·1633	9·50	2·2513
7·12	1·9629	7·92	2·0694	8·72	2·1656	9·52	2·2534
7·14	1·9657	7·94	2·0719	8·74	2·1679	9·54	2·2555
7·16	1·9685	7·96	2·0744	8·76	2·1702	9·56	2·2576
7·18	1·9713	7·98	2·0769	8·78	2·1725	9·58	2·2596
7·20	1·9741	8·00	2·0794	8·80	2·1747	9·60	2·2617
7·22	1·9768	8·02	2·0819	8·82	2·1770	9·62	2·2638
7·24	1·9796	8·04	2·0844	8·84	2·1793	9·64	2·2659
7·26	1·9823	8·06	2·0869	8·86	2·1815	9·66	2·2680
7·28	1·9851	8·08	2·0894	8·88	2·1838	9·68	2·2700
7·30	1·9878	8·10	2·0918	8·90	2·1860	9·70	2·2721
7·32	1·9906	8·12	2·0943	8·92	2·1883	9·72	2·2742
7·34	1·9933	8·14	2·0968	8·94	2·1905	9·74	2·2762
7·36	1·9660	8·16	2·0992	8·96	2·1927	9·76	2·2783
7·38	1·9987	8·18	2·1017	8·98	2·1950	9·78	2·2803
7·40	2·0015	8·20	2·1041	9·00	2·1972	9·80	2·2824
7·42	2·0041	8·22	2·1065	9·02	2·1994	9·82	2·2844
7·44	2·0068	8·24	2·1090	9·04	2·2016	9·84	2·2864
7·46	2·0095	8·26	2·1114	9·06	2·2038	9·86	2·2885
7·48	2·0122	8·28	2·1138	9·08	2·2060	9·88	2·2905
7·50	2·0149	8·30	2·1162	9·10	2·2082	9·90	2·2925
7·52	2·0176	8·32	2·1186	9·12	2·2104	9·92	2·2945
7·54	2·0202	8·34	2·1210	9·14	2·2126	9·94	2·2965
7·56	2·0228	8·36	2·1234	9·16	2·2148	9·96	2·2985
7·58	2·0255	8·38	2·1258	9·18	2·2170	9·98	2·3006
7·60	2·0281	8·40	2·1282	9·20	2·2192	10·0	2·3026
7·62	2·0307	8·42	2·1306	9·22	2·2213	10·2	2·322
7·64	2·0333	8·44	2·1330	9·24	2·2235	10·4	2·342
7·66	2·0360	8·46	2·1353	9·26	2·2257	10·6	2·361
7·68	2·0386	8·48	2·1377	9·28	2·2278	10·8	2·379
7·70	2·0412	8·50	2·1400	9·30	2·2300	11·0	2·398
7·72	2·0438	8·52	2·1424	9·32	2·2321	11·2	2·416
7·74	2·0464	8·54	2·1447	9·34	2·2343	11·4	2·433
7·76	2·0490	8·56	2·1471	9·36	2·2364	11·6	2·451
7·78	2·0515	8·58	2·1494	9·38	2·2386	11·8	2·468
7·80	9·0541	8·60	2·1517	9·40	2·2407	12·0	2·485

No.	Logarithm.	No.	Logarithm.	No.	Logarithm.	No.	Logarithm
12·2	2·501	20·2	3·006	41·	3·713	81·	4·394
12·4	2·518	20·4	3·016	42·	3·737	82·	4·406
12·6	2·534	20·6	3·026	43·	3·761	83·	4·418
12·8	2·549	20·8	3·035	44·	3·784	84·	4·431
13·0	2·565	21·0	3·044	45·	3·806	85·	4·442
13·2	2·580	21·2	3·053	46·	3·828	86·	4·454
13·4	2·595	21·4	3·062	47·	3·850	87·	4·466
13·6	2·610	21·6	3·071	48·	3·871	88·	4·477
13·8	2·625	21·8	3·080	49·	3·892	89·	4·488
14·0	2·639	22·0	3·091	50·	3·912	90·	4·500
14·2	2·653	22·2	3·100	51·	3·932	91·	4·511
14·4	2·667	22·4	3·109	52·	3·951	92·	4·522
14·6	2·681	22·6	3·118	53·	3·970	93·	4·532
14·8	2·694	22·8	3·127	54·	3·989	94·	4·543
15·0	2·708	23·0	3·135	55·	4·007	95·	4·554
15·2	2·721	23·2	3·144	56·	4·025	99·	4·564
15·4	2·734	23·4	3·153	57·	4·043	99·	4·574
15·6	2·747	23·6	3·162	58·	4·060	99·	4·585
15·8	2·760	23·8	3·170	59·	4·077	99·	4·595
16·0	2·772	24·0	3·178	60·	4·094	100·	4·605
16·2	2·785	24·2	3·186	61·	4·111	150·	5·010
16·4	2·797	24·4	3·194	62·	4·127	200·	5·298
16·6	2·809	24·6	3·202	63·	4·143	250·	5·521
16·8	2·821	24·8	3·211	64·	4·159	300·	5·704
17·0	2·833	25·0	3·219	65·	4·174	350·	5·858
17·2	2·845	26·	3·258	66·	4·190	400·	5·991
17·4	2·857	27·	3·296	67·	4·205	450·	6·109
17·6	2·868	28·	3·332	68·	4·219	500·	6·215
17·8	2·879	22·	3·367	69·	4·234	600·	6·397
18·0	2·890	30·	3·401	70·	4·248	700·	6·551
18·2	2·901	31·	3·434	71·	4·263	800·	6·685
18·4	2·912	32·	3·465	72·	4·277	900·	6·802
18·6	2·923	33·	3·496	73·	4·290	1000·	6·908
18·8	2·934	34·	3·526	74·	4·304	5000·	8·517
19·0	2·945	35·	3·555	75·	4·317	10000·	9·210
19·2	2·955	36·	3·583	76·	4·330		
19·4	2·965	37·	3·611	77·	4·[illegible]4		
19·6	2·975	38·	3·637	78·	4·357		
19·8	2·986	39·	3·663	79·	4·369		
20·0	2·996	40·	3·689	80·	4·382		

When the expansion of steam, or of any other gaseous fluid, follows the law of MARRIOTTE, that is, when its pressure during expansion is in the inverse ratio of its volume, the line of pressure thus resulting is designated the "Hyperbolic Curve." The Hyperbolic Logarithm represents the value of the area produced by such curve, in proportion to unity of area produced to the point where the curve of expansion commences. For illustration let us take the theoretical expansion diagram on page 79, where the cut-off is at one-tenth, so that the volume at the termination of the expansion is ten times the initial volume, and the pressure is one-tenth of the initial pressure. To calculate the value of the expansion, the hyperbolic logarithm of 10 will be required. It will be seen by the table that the hyperbolic logarithm of 10 is 2·3026 (the exact figures are 2·302585, but those usually given are 2·303). If the area produced by the steam to the point of cut-off be denoted by 1·, then the area produced by expansion will be 2·3026, and therefore that of the whole diagram will be 3·3026. The reader will understand that the area is exactly proportionate to the power.

In order to ascertain the average pressure of a diagram having the hyperbolic expansion curve, multiply the absolute terminal pressure by the hyperbolic logarithm plus 1· of the number representing the range of expansion. Thus the theoretical diagram just referred to has a range of expansion of 10; and as the terminal pressure is 7·5lbs, this must be multiplied by (2·3026 + 1 =) 3·3026, which will give an average pressure of 24·7695lbs absolute. If the cut-off should occur at half stroke, the steam would expand to two volumes, and the value of expansion would be represented by the hyperbolic logarithm of 2, which is 0·693. Suppose now that the cut-off should occur at three-tenths of the stroke, then it would expand from three to ten volumes, and the range of expansion would be 3·333. The hyperbolic logarithm of 3·333 is 1·203, and if the absolute terminal pressure be multiplied by 1·203 + 1 (= 2·203) the average absolute pressure will be found. The table does not give the number 3·33, but the hyperbolic logarithm of it will be obtained with sufficient accuracy by taking the mean of 3·32 and 3·34. If the cut-off should occur at two-thirds of the stroke, then the range of expansion will be 1·5, the hyperbolic logarithm of which is 0·4054. To this must be added 1, making 1·4054, with which the terminal absolute pressure is to be multiplied to obtain the average absolute pressure of whole diagram.

TABLE V.

CO-EFFICIENTS OF LINEAR EXPANSION BY HEAT.

Gold, Paris standard,	1° Centigrade.	1° Fahrenheit.	100° Centigrade 180° Fahrenheit
annealed ...	0·000015153	0·000008418	0·0015153
„ unannealed	0·000015515	0·000008620	0·0015515
Silver	0·000019086	0·000010603	0·0019086
Copper.	0·000017173	0·000009540	0·0017173
Tin, Falmouth ...	0·000021729	0·000012070	0·0021729
„ Malacca ...	0·000019376	0·000010765	0·0019376
Zinc	0·000029400	0·000016333	0·0029400
Brass (by Rankine)	0·000021600	0·000012000	0·0021600
„ (by Deschanel)	0·000018782	0·000010434	0·0018782
Lead	0·000029000	0·000016111	0·0029000
Iron, soft, wrought ...	0·000012204	0·000006780	0·0012204
„ cast	0·000011000	0·000006111	0·0011000
Steel, soft	0·000010792	0·000005995	0·0010792
„ tempered and re-heated to 65°	0·000012395	0·000006886	0·0012395
Platinum	0·000009918	0·000005510	0·0009918
Glass, flint	0·000008116	0·000004509	0·0008116
„ French, with lead	0·000008715	0·000004841	0·0008715
Bismuth	0·000013916	0·000007731	0·0013916

TABLE VI.—EXPANSION OF VOLUME.

	1° Centigrade	1° Fahrenheit
Mercury ... ... 32° to 212° Fahrenheit	0·00018000	0·00010000
„ ... 212° „ 392° „	0·00018450	0·00010250
„ ... ... 392° „ 572° „	0·00018900	0·00010500
Water ... ... 32° „ 212° „	0·00047556	0·00026420
„ ... ... 212° „ 392° „	0·00091836	0·00051020
„ ... ... 392° „ 572° „	0·00102083	0·00056713
Air, and all permanent Gases 32° „ 212° „	0·00366300	0·00203500

To find the amount of expansion for any given range of temperature, the co-efficient given must be multiplied by the number of degrees through which the body may have passed. The third column of Co-efficients of Linear Expansion represents the proportion of their

whole length which the several bodies will expand, by a change of temperature from the freezing to the boiling point of water at atmospheric pressure. Liquids are more conveniently and more correctly represented by their increase of volume. To change the decimal to the vulgar fraction may enable some persons to see the proportions better. This can be very easily done. Take the last co-efficient as an example. Now there are six places of decimals necessary to be taken into account—the cyphers to the right hand may be entirely disregarded; after these place seven figures, and cut off the two cyphers to the left hand thus: 0·00 | 2035) 1000000 (491·4; so that the expansion is 1 — 491·4 of its volume for 1° Fahrenheit.

As another example, take the first co-efficient of linear expansion, which is for one degree centigrade for gold. Here are nine places of decimals, thus: 0·0000 | 15153) 1,000,000,000 (65993; so that the amount of expansion is $\frac{1}{65993}$ of its length for one degree centigrade increase of its temperature. For one degree Fahrenheit it is obvious that the expansion would be still less in the ratio of 9 to 5, which would make it $\frac{1}{118788}$ of its length. The co-efficient for air, &c., is usually allowed to remain constant through all temperatures.

Table VII.—CONDUCTIVITY OF METALS.

As given by Tyndall.

	For Heat.	For Electricity
Silver	100	100
Copper	74	73
Gold	53	59
Brass	24	22
Tin	15	23
Iron	12	13
Lead	9	11
Platinum	8	10
German Silver	6	6
Bismuth	2	2

As given by Deschanel.

	For Heat.
Silver	100
Copper	77·6
Gold	53·2
Brass	33·
Zinc	19·9
Tin	14·5
Steel	12·
Iron	11·9
Lead	8·5
Platinum	8·2
Palladium	6·3
Bismuth	1·9

As Silver has the highest conductivity of any metal, it is taken as the standard by which to exhibit the comparative values of all others.

TABLE VIII.

TEMPERATURES OF FUSION.

	Centigrade	Fahrenheit
Mercury	−40	−40
Bromine	−7·32	+18·824
Ice	0	+32
Butter	+33	91·4
Lard	33	91·4
Spermaceti	49	120·2
Stearine	55	131
Yellow Wax	62	143·6
White "	68	154·4
Stearic Acid	70	158
Phosphorus	44	111·2
Potassium	63	165·4
Sodium	95	203
Iodine	107	224·6
Sulphur	110	230
Tin	230	446
Cadmium	315	599
Bismuth	265	509
Lead	320	608
Zinc	360	680
Antimony	432	810
Bronze	900	1652
Silver, pure	1000	1832
Copper	1150	2102
Gold, coined	1180	2156
Gold, pure	1250	2282
Iron, cast	1050	1922
	1250	2282
Iron, wrought	1500	2732
	1600	2912
Steel	1300	2372
	1400	2552
Platinum	2000	3632
Brass	1020	1868
Alloy—Tin 4, Bismuth 5, Lead 1	119	246

TABLE IX.

BOILING POINTS, OR TEMPERATURES OF EVAPORATION OF THE FOLLOWING LIQUIDS AT ATMOSPHERIC PRESSURE.

	Centigrade	Fahrenheit
Sulphuric Acid	−10	+14
Hydrochloric Ether	+11	51·8
Common "	37	98·6
Alcohol	79	174·2
Distilled Water	100	212
Spirits of Turpentine	130	266
Phosphorus	290	554
Concentrated Sulphuric Acid	325	617
Mercury	353	667·4
Sulphur	440	824

Table X.—HEATS OF COMBUSTION.

	Centigrade	Fahrenheit
Hydrogen	34462	62032
Hydrogen with Chlorine	23783	42809
Marsh Gas	13063	23513
Olefiant Gas	11858	21344
Oil of Turpentine	10852	19534
Olive Oil	9862	17752
Stearic Acid	9616	17309
Ether	9028	16250
Carbon completely burned, to Carbonic Acid	8055·5	14500
Carbon incompletely burned, to Carbonic Oxide	2444·4	4400
Carbonic Oxide, the product of 1lb of Carbon	5611·1	10100
Carbonic Oxide, unit of weight	2404	4328
Charcoal	5700	13500
Graphite	7797	14035
Diamond	7770	13986
Alcohol	7184	12931
Native Sulphur	2261	4070
Soft Sulphur	2258	4064
Sulphide of Carbon	3400	6120

The numbers represent the number of times its own weight of water which would *be raised* 1° by the heat evolved by the complete combustion of the substance.

TABLE XI.

TOTAL HEAT OF COMBUSTION OF FUEL.

FUEL.	Carbon.	Hydrogen	Oxygen	Heat equivlnt. to 1 in weight of pure Carbon.	Pounds of Water evporatd. at 212° by 1 lb fuel.	Heat in British Thermal Units.	Air requir'd to one of Fuel, in weight.
Charcoal, from wood	0·93			0·93	14	13500	11·16
„ „ peat				0·80	12	11600	9·6
Coke, good	0·94			0·94	14	13620	11·28
„ middling	0·88			0·88	13·2	12760	10·56
„ bad	0·82			0·82	12·3	11890	9·84
Coal:							
1 Anthracite	0·915	0·035	0·026	1·05	15·75	15225	12·13
2 Dry Bituminous	0·90	0·04	0·02	1·06	15·9	15370	12·06
3 „ „	0·87	0·04	0·03	1·025	15·4	14860	12·6
4 „ „	0·80	0·054	0·016	1·02	15·3	14790	12·24
5 „ „	0·77	0·05	0·06	0·95	14·25	13775	11·4
6 Caking	0·88	0·052	0·054	1·075	16·0	15837	12·9
7 „	0·81	0·052	0·04	1·01	15·15	14645	12·12
8 Cannel	0·84	0·056	0·08	1·04	15·6	15080	12·48
9 Dry long flaming	0·77	0·052	0·15	0·91	13·65	13195	10·92
10 Lignite	0·70	0·05	0·20	0·81	12·15	11745	9·72
Peat, dry	0·58	0·06	0·31	0·66	10·0	9660	7·92
„ 25 % moisture					7·25	7000	5·74
Wood, dry	0·50			0·50	7·5	7245	5·94
„ 25 % moisture					5·8	5600	4·63

The figures in the fifth column represent the number of pounds of water evaporated by one pound of the substance, on the assumption that the combustion is complete, and that all the heat evolved is transmitted to the water. The initial temperature of the water is understood to be at 212° Fahrenheit, and is evaporated at the same temperature. This represents 966 British thermal units per pound of water; so that taking pure carbon, which is equal to 14,500 thermal units per pound avoirdupois, it will be equivalent to fifteen pounds of water evaporated from and at 212°, or the boiling point at atmospheric pressure.

TABLE XII.

SPECIFIC HEATS.

Water................................ 1·000.

SOLIDS.

Antimony	0·05077	Brass	0·09391
Silver...	0·05601	Nickel...	0·10860
Arsenic	0·08140	Gold	0·03244
Bismuth	0·03084	Phosphorous	0·18870
Cadmium	0·05669	Platinum	0·03243
Charcoal	0·24150	Lead	0·03140
Copper...	0·09215	Plumbago	0·21800
Diamond	0·14680	Sulphur	0·20259
Tin	0·05623	Glass	0·19768
Iron	0·11379	Zinc	0·09555
Iodine	0·05412	Ice...	0·50400

LIQUIDS.

Mercury	0.03332	Benzine	0·39520
Acetic Acid	0·65890	Ether	0·51570
Acohol at 36° centigrade	0·67350	Oil of Turpentine ...	0·46290
Olive Oil	0·30960	Petroleum	0·46840

GASES AT CONSTANT PRESSURE.

Air	0·2380	Carbonic Acid	0·2170
Oxygen...	0·2180	Olefiant Gas... ...	0·3690
Hydrogen	3·4050	Nitrogen	0·2440
Steam	0·4750	Chlorine	0·1214
Ether Vapour ...	0·4810	Ammonia	0·5080

The "Specific Heat" of any body is its capacity for heat, that is, the quantity of heat required to raise a unit of its weight through one degree in temperature; otherwise designated "heat capacity." Whatever values the units of weight and of heat may be fixed at, the proportions of specific heat shown in the table will remain the same. Pure water at its greatest density, which is 39·1° Fahrenheit, and with a limited range of temperature upwards, has been adopted as the unit of specific heat. One pound avoirdupois weight of water *raised one* degree Fahrenheit represents a British thermal unit;

and if expressed by the centigrade scale, would be equivalent to 0·555°. The French thermal unit of heat is equivalent to one kilogramme of water raised in temperature one degree centigrade. As a kilogramme equals 2·20462lbs avoirdupois, and as 1° centigrade equals 1·8° Fahrenheit, then a French thermal unit will equal 3·9683 British thermal units.

British thermal unit = 1lb water raised 1° Fahrenheit.
French thermal unit = 1 kilogramme of water raised 1° centigrade.

The mechanical equivalent of heat is 772 foot pounds for one British thermal unit. For 1° centigrade it is 772 × 1·8 = 1389·6 foot pounds.

The specific heat of the gases is given for constant pressure, so that a portion of the absorbed heat is expended in overcoming a resisting weight or pressure, rendered necessary by the increase of volume. Heat is thus imparted, in excess of the sensible heat which is shown by the thermometer, equivalent to the work performed. The proportions of specific heat at constant volume and at constant pressure, of simple gases, and of air, which follows the same law, are approximately as 1 : 1·4; so that if the specific heat given for constant pressure be divided by 1·4, that of constant volume will be obtained.

Analogy leads to the conclusion, which is now accepted as an established law, that liquids and solids would also have a lower specific heat if, by sufficient force of resistance, they could be maintained at the same volume and dimensions whilst receiving increments of temperature. The heat absorbed in excess of the quantity required by the increase of temperature under the conditions here supposed, may be called the *latent heat of expansion.*

TABLE XIII.

SPECIFIC GRAVITIES.

METALS.

Iridium	21·8	Iron	7·8
Platinum	21·5	Tin	7·3
Gold	19·3	Zinc	7·1
Mercury	13·596	Antimony	6·7
Thallium	11·9	Arsenic	5·9
Palladium	11·8	Chromium	5·9
Lead	11·3	Aluminium	2·56
Silver	10·5	Strontium	2·54
Bismuth	9·8	Magnesium	1·75
Copper	8·9	Calcium	1·58
Nickel	8·8	Rubidium	1·52
Cadmium	8·6	Sodium	0·972
Cobalt	8·5	Potassium	0·865
Manganese	8·0	Lithium	0·593

OTHER SOLIDS.

Basalt	3·0	Chalk	1·8 to 2·8
Clay	1·92	Sulphur	2·
Glass, Crown	2·5	Granite, Aberdeen	2·625
„ Flint	3·0	Limestone, Green	3·182
Quartz, Rock Crystal	2·65	Marble, Parian	2·838
Portland Stone	2·57	Porphyry, Red	2·765
Talc, Black	2·9	Slate	2·672
Pumice Stone	0·915	Ice	0·92

LIQUIDS.

Water, Pure	1·000	Oil of Turpentine	0·870
„ Sea, ordinary	1·026	Blood, Human	1·055
Alcohol, Pure	0·791	Milk, Cow	1·03
„ Proof Spirit	0·916	Acetic Acid	1·007
Ether	0·716	Ammonia, Liquid	0·900
Naptha	0·848	„ Muriate of	1·45
Oil, Linseed	0·940	Turpentine	0·990
„ Olive	0·915	Sulphuric Acid	1·840
„ *Whale*	0·923	„ „ concentrtd.	2·125

GASES.

Atmospheric Air ... 1·000,—being to Water as 1 : 773·28.
Air being at 32° and Water at 39·1° Fahrenheit.

Hydrogen	0·069	Carburetted Hydrogen	0·972
Oxygen	1·104	Sulphuretted Hydrogen	1·777
Nitrogen	0·972	Chlorine	2.500
Carbonic Acid	1·527	Ammonical	0·500
„ Oxide	0·972	Vapour of Alcohol ...	1·613

TABLE XIV.

HEIGHTS DUE TO VELOCITIES PRODUCED BY GRAVITY.

Velocity in feet ℔ Second	Height.	Velocity in feet ℔ Second	Height.	Velocity in feet ℔ Second	Height.
1	0·01553	27	11·320	54	45·279
2	0·06211	28	12·174	56	48·695
3	0·13975	29	13·059	58	52·235
4	0·24844	30	13·975	60	55·870
5	0·38819	31	14·922	62	59·689
6	0·55900	32	15·900	64	63·602
7	0·76086	32·2	16·100	64·4	64·400
8	0·99377	33	16·910	66	67·639
9	1·2577	34	17·950	68	71·800
10	1·5528	35	19·021	70	76·087
11	1·8789	36	19·813	72	80·497
12	2·2360	37	21·257	74	85·029
13	2·6241	38	22·422	76	89·688
14	3·0434	29	23·618	78	94·471
15	3·4937	40	24·844	80	99·379
16	3·9751	41	26·102	82	104·41
17	4·4875	42	27·391	84	109·56
18	5·0310	43	28·711	86	114·84
19	5·6055	44	30·062	88	120·25
20	6·2111	45	31·444	90	125·77
21	6·8477	46	32·857	92	131·43
22	7·5153	47	34·301	94	137·20
23	8·2141	48	35·776	96	143·10
24	8·9441	49	37·282	98	149·07
25	9·7048	50	38·820	100	155·28
26	10·497	52	41·987		

The height is found by squaring the velocity and dividing by 64·4, and the velocity is found by reversing the process.

TABLE XV.

By Act of Parliament (27 and 28 Vict., cap. 117, 29th July, 1864) the use of the Metrical System of Weights and Measures is rendered legal. The weight of the Kilogramme is settled by this Act to be 15432·3487 English grains.

COMPARISON OF THE METRICAL WITH THE COMMON MEASURES. By Dr. Warren De La Rue.

MEASURES OF LENGTH.

	In English Inches.	In English Feet =12 Inches.	In English Yards =3 Feet.	In English Fathoms =6 Feet.	In English Miles =1,760 Yards.
Millimetre	0·03937	0·0032809	0·0010936	0·0005468	0·0000006
Centimetre	0·39371	0·0328090	0·0109363	0·0054682	0·0000062
Decimetre	3·93708	0·3280899	0·1093633	0·0546816	0·0000621
Metre	39·37079	3·2808992	1·0936331	0·5468165	0·0006214
Decametre	393·70790	32·8089920	10·9363310	5·4681655	0·0062138
Hectometre	3937·07900	328·0899200	109·3633100	54·6816550	0·0621882
Kilometre	39370·79000	3280·8992000	1093·6331000	546·8165500	0.6213824
Myriometre	393707·90000	32808·9920000	10936·3310000	5468·1655000	6·2138244

1 Inch = 2·539954 Centimetres.
1 Foot = 3·0479449 Decimetres.
1 Yard = 0·91438348 Metre.
1 Mile = 1·6093149 Kilometre.

MEASURES OF SURFACE.

	In English Square Feet.	In English Sq. Yards = 9 Square Feet.	In English Poles = 272·25 Sq. Feet.	In English Roods = 10,890 Sq. Feet.	In English Acres = 43,560 Sq. Feet.
Centare or sq. metre	10·7642993	1·1960333	0·0395383	0·000988457	0·0002471148
Are or 100 sq. metres	1076·4299342	119·6033260	3·9538290	0·098845724	0·0247114810
Hectare or 10,000 sq. metres ...	107642·9934183	11960·3326020	395·3828959	9·884572398	2·4711430996

1 Square Inch = 6·4513669 Square Centimetres.
1 Square Foot = 9·2899683 Square Decimetres.
1 Square Yard = 0·83609715 Square Metre or Centare.
1 Acre = 0·404671021 Hectare.

MEASURES OF CAPACITY.

	In Cubic Inches.	In Cubic Feet = 1,728 Cubic Inches.	In Pints = 34·65923 Cubic Inches.	In Gallons = 8 Pints = 277·27384 Cubic Inches.	In Bushels = 8 Gallons = 2218·19075 Cubic Inches.
Millilitre, or cubic centimetre ...	0·061027	0·0000353	0·001761	0·00022010	0·000027512
Centilitre, or 10 cubic centimetres	0·610271	0·0003532	0·017608	0·00220097	0·000275121
Decilitre, or 100 cubic centimetres	6·102705	0·0035317	0·176077	0·02200967	0·002751208
Litre, or cubic decimetre	61·027052	0·0353166	1.760773	0·22009668	0·027512085
Decalitre, or centistere	610·270515	0·3531658	17·607734	2·20096677	0·275120846
Hectolitre, or decistere	6102·705152	3·5316581	176·077341	22·00966767	2·751208459
Kilolitre, or stere, or cubic metre .	61027.051519	35·3165807	1760·773414	220·09667675	27·512084594
Myriolitre, or decastere	610270·515194	353·1658074	17607·734140	2200·96676750	275·120845937

1 Cubic Inch = 16·3861759 Cubic Centimetres. 1 Cubic Foot = 28·3153119 Cubic Decimetres. 1 Gallon = 4·543457969 Litres.

MEASURES OF WEIGHT.

	In English Grains.	In Troy Ounces = 480 Grains.	In Avoirdupois lbs. = 7,000 Grains.	In Cwts. = 112 lbs. = 734,000 Grains.	In Tons = 20 Cwt. = 15,620,000 Grains.
Milligramme	0·015432	0·000032	0·0000022	0·00000002	0·000000001
Centigramme	0·154323	0·000322	0·0000220	0·00000020	0·000000010
Decigramme	1·543235	0·003215	0·0002205	0·00000197	0·000000098
Gramme	15·432349	0·032151	0·0022046	0·00001968	0·000000984
Decagramme	154·323488	0·321507	0·0220462	0·00019684	0·000009842
Hectogramme	1543·234880	3·215073	0·2204621	0·00196841	0·000098421
Kilogramme	15432·348800	32·150727	2·2046213	0·01968412	0·000984206
Myriogramme	154323·488000	321·507267	22·0462126	0·19684118	0·009842059

1 Grain = 0·064798950 Gramme.
1 Troy oz. = 31·103496 Gramme.
1lb Avd. = 0·45359265 Kilogr.
1 Cwt. = 50·80237689 Kilogr.

TABLE XVI.

SHOWING THE INCREASE AND DECREASE OF THE STRENGTH OF IRON AT VARIOUS TEMPERATURES, FROM TESTS BY THE COMMITTEE OF THE FRANKLIN INSTITUTE.

Comparative view of the influence of High Temperature on the strength of Iron, as exhibited by 73 experiments on 47 different specimens of that metal, at 40 different temperatures, from 212° to 1317° Fahrenheit, compared with the strength of each bar when tried at ordinary temperatures : the whole number of experiments at the latter being 163.

No. of the comparison.	Temperature observed at the moment of fracture	Mark of the Bar on which the trial was made.	Strength at ordinary temperatures.	No. of experiments at ordinary temperatures.	Strength at the temperature observed.	No. of experiments at high temperatures.	Amount of variation from uniformity in the cold experiments.	Effects of the heat, expressed in parts of the original strength.	REMARKS.
1	212°	137	56736	1	67939	1		+·198	
2	214	133	53176	1	61161	1		+·150	
3	394	58	68356	1	71896	1		+·052	
4	394	148	65143	1	69752	1		+·070	
5	394	23	62646	2	67765	1	·1041	+·081	
6	394	125	57182	1	63322	1		+·107	
7	394	61	55297	5	61917	1	·2026	+·119	
8	396	75	60433	3	62415	1	·0444	+·031	
9	440	224D	49782	4	59085	1	·0908	+·187	
10	520	224B	54934	4	58451	1	·0992	+·064	
11	550	199A	76986	4	79846	2	·0936	+·037	The standard for the original strength may possibly be a little too high
12	550	221A	60518	4	60322	1	·1680	—·003	
13	552	14	52542	1	55932	1		+·064	
14	554	218A	58124	4	60412	1	·0730	+·039	
15	554	22	54372	4	61680	3	·1919	+·134	
16	560	224E	50528	7	58824	1	·0605	+·158	
17	562	224C	53385	5	59623	1	·1919	+·104	
18	563	60	60907	4	72588	2	·0460	+·191	
19	564	74	51030	5	58824	1	·0764	+·142	
20	568	9	67211	2	76763	1	·0601	+·042	Standa. probably too high for the mean strength.
21	572	219B	66724	2	66620	1	·0325	—·002	
22	572	49	59607	3	62278	1	·0878	+·045	
23	572	222B	56165	4	60117	2	·1550	+·070	
24	573	10	64511	1	67503	3		+·046	
25	574	231	76071	5	65387	1	·1373	+·014	
26	575	220A	54263	4	60988	1	·0280	+·124	
27	575	62	58376	3	70081	3	·0262	+·200	
28	575	207	51924	5	63825	3	·1225	+·229	
29	576	221B	59234	5	66065	1	·1190	+·115	
30	576	223B	43386	6	50068	1	·0760	+·154	

No. of the comparison.	Temperature observed at the moment of fracture.	Mark of the bar on which the trial was made.	Strength at ordinary temperatures.	No. of exps. at ord. temps.	Strength at the temperature observed.	No. of exps at high temps.	Amount of variation from uniformity in the cold experiments.	Effects of the heat, expressed in parts of the original strength.	REMARKS.
31	577°	164	58769	5	66929	2	·1214	+·139	
32	578	224A	52406	5	59197	1	·0565	+·129	
33	578	223A	45757	5	53465	1	·0896	+·168	
34	580	86	62156	3	77163	2	·0986	+·052	
35	590	220B	52459	5	62966	1	·0680	+·058	
36	598	90	50316	5	57310	2	·2401	+·138	
37	·630	219A	59530	3	60010	1	·1440	+·008	
38	636	16	53543	1	50039	1	·1563	—·067	This experiment was on a part probably defective.
39	662	150	59307	5	58181	1	·0644	—·019	
40	722	152	57133	3	54441	1	·0507	—·047	
41	732	14	52542	1	53378	1	·1310	+·016	
42	734	150	59397	1	57903	1	·0644	—·026	
43	766	16	56891	1	54819	1	·1563	—·037	
44	770	149	56825	2	54781	1	·0234	—·036	
45	814 and 824	214	59219	1	55892	1	·0413	—·073	
46	825	149	56825	2	56644	1	·0234	—·029	
47	932	214	59219	1	45531	1	·0413	—·240	
48	947	232	58341	2	42401	1	·0446	—·273	
49	1022	214	59219	2	37410	1	·0413	—·369	
50	1037	152	58992	1	37764	1	·0507	—·360	
51	1097	227	53426	6	27604	1	·0330	—·483	
52	1111	227	53426	6	27602	1	·0330	—·483	
53	1142	226	54758	2	18672	1	·1147	—·659	The metal was decidedly defective at the point where this fracture was made—flaws visible.
54	1155	227	53426	6	21967	1	·0330	—·589	
55	1159	229	55774	3	25620	1	·1102	—·538	The 8th experiment on this bar being taken as the standard would exhibit the effect—·550.
56	1187	227	53426	6	21910	1	·0330	—·589	
57	1235	226	54758	2	21298	1	·1147	—·611	
58	1245	226	54758	2	20703	1	·1147	—·622	
59	1317	226	54758	2	18913	1	·1147	—·654	

Mean=57525

This table shows that Iron Steam Boilers become stronger as the temperature increases, up to about 600 degrees; beyond that temperature it becomes weaker.

Copper decreases in strength from 56 degrees upwards.

TABLE XVII.

CONTAINING THE WEIGHT OF ROUND BAR IRON, FROM 1 TO 10 FEET IN LENGTH, AND FROM ¼ INCH TO 6 INCHES DIAMETER.

Inches Diameter	LENGTH OF THE BARS IN FEET.									
	1 foot	2 feet	3 feet	4 feet	5 feet	6 feet	7 feet	8 feet	9 feet	10 feet
	lbs	lbs	lbs	lbs	lbs	lbs	lbs	lbs	lbs	lbs
¼	0·2	0·3	0·5	0·7	0·8	1·0	1·2	1·3	1·5	1·7
⅜	0·4	0·7	1·1	1·5	1·9	2·2	2·6	3·0	3·4	3·7
½	0·7	1·3	2·0	2·7	3·3	4·0	4·6	5·3	6·0	6·6
⅝	1·0	2·1	3·1	4·2	5·2	6·3	7·3	8·3	9·4	10·4
¾	1·5	3·0	4·5	6·0	7·5	9·0	10·5	11·9	13·4	14·9
⅞	2·0	4·1	6·1	8·1	10·2	12·2	14·2	16·3	18·3	20·3
1 in.	2·7	5·3	8·0	10·6	13·3	15·9	18·6	21·2	23·9	26·5
1⅛	3·4	6·7	10·1	13·4	16·8	20·2	23·5	26·9	30·2	33·6
1¼	4·2	8·3	12·5	16·7	20·9	25·0	29·2	33·4	37·5	41·7
1⅜	5·0	10·0	15·1	20·1	25·1	30·1	35·1	40·2	45·2	50·2
1½	6·0	11·9	17·9	23·9	29·9	35·8	41·8	47·8	53·7	59·7
1⅝	7·0	14·0	21·0	28·0	35·1	42·1	49·1	56·1	63·1	70·1
1¾	8·1	16·3	24·4	32·5	40·6	48·8	56·9	65·0	73·2	81·3
1⅞	9·3	18·7	28·0	37·3	46·7	56·0	65·3	74·7	84·0	93·3
2 in.	10·6	21·2	31·8	42·5	53·1	63·7	74·3	84·9	95·5	106·2
2⅛	12·0	24·0	36·0	48·0	59·9	71·9	83·9	95·9	107·9	119·9
2¼	13·4	26·9	40·3	53·8	67·2	80·6	94·1	107·5	121·0	134·4
2⅜	15·0	30·0	45·0	60·0	75·0	90·0	105·0	120·0	135·0	150·0
2½	16·7	33·4	50·1	66·8	83·4	100·1	116·8	133·5	150·2	166·9
2⅝	18·3	36·6	54·9	73·2	91·5	109·8	128·1	146·3	164·6	182·9
2¾	20·1	40·2	60·2	80·3	100·4	120·5	140·5	160·6	180·7	200·8
2⅞	21·9	43·9	65·8	87·8	109·7	131·7	153·6	175·6	197·5	219·4
3 in.	23·9	47·8	71·7	95·6	119·4	143·3	167·2	191·1	215·0	238·9
3⅛	25·9	51·9	77·8	103·7	129·6	155·6	181·5	207·4	233·3	259·3
3¼	28·0	56·1	84·1	112·2	140·2	168·2	196·3	224·3	253·4	280·4
3⅜	30·2	60·5	90·7	121·0	151·2	181·4	211·7	241·9	272·2	302·4
3½	32·5	65·0	97·5	130·0	162·6	195·1	227·6	260·1	292·6	325·1
3⅝	34·9	69·8	104·7	139·5	174·4	209·3	244·2	279·1	314·0	348·9
3¾	37·3	74·7	112·0	149·3	186·7	224·0	261·3	298·7	336·0	373·3
3⅞	39·9	79·7	119·6	159·5	199·3	239·2	279·0	318·9	358·8	398·6
4 in.	42·5	84·9	127·4	169·9	212·3	254·8	297·2	339·7	382·2	424·6
4⅛	45·2	90·3	135·5	180·7	225·9	271·0	316·2	361·4	406·6	451·7
4¼	48·0	95·9	143·9	191·8	239·8	287·7	335·7	383·6	431·6	479·5
4⅜	50·8	101·6	152·4	203·3	254·1	304·9	355·7	406·5	457·3	508·2
4½	53·8	107·5	161·3	215·0	268·8	322·6	376·3	430·1	483·8	537·6
4⅝	56·8	113·6	170·4	227·2	283·9	340·7	397·5	454·3	511·1	567·9
4¾	60·0	119·8	179·7	239·6	299·5	359·4	419·3	479·2	539·1	599·0
4⅞	63·1	126·2	189·3	252·4	315·5	378·6	441·7	504·8	567·8	630·9
5 in.	66·8	133·5	200·3	267·0	333·8	400·5	467·3	534·0	600·8	667·5
5¼	73·2	146·3	219·5	292·7	365·9	439·0	512·2	585·4	658·5	731·7
5½	80·3	160·6	240·9	321·2	401·5	481·8	562·1	642·4	722·7	803·0
5¾	87·8	175·6	263·3	351·1	438·9	526·7	614·4	702·2	790·0	877·8
6 in.	95·6	191·1	286·7	382·2	477·8	573·3	668·9	764·4	860·0	955·5

TABLE XVIII.

CONTAINING THE WEIGHT OF CAST-IRON BALLS FROM THREE TO TWELVE INCHES DIAMETER.

Diameter in Inches.	Weight in Pounds.	Diameter in Inches.	Weight in Pounds.	Diameter in Inches.	Weight in Pounds.
3	3·7	6	29·7	9	100·3
3¼	4·7	6¼	33·6	9¼	108·9
3½	5·8	6½	37·8	9½	118·0
3¾	7·2	6¾	42·3	9¾	127·6
4	8·8	7	47·2	10	137·7
4¼	10·5	7¼	52·4	10¼	148·2
4½	12·5	7½	58·0	10½	159·4
4¾	14·7	7¾	64·0	10¾	171·0
5	17·1	8	70·4	11	183·2
5¼	19·9	8¼	77·3	11½	209·4
5½	22·9	8½	84·5	12	237·9
5¾	26·1	8¾	92·2		

To find the weight of a sphere of Cast Iron, cube the diameter in inches, and divide by 7·27. The quotient will be the weight in lbs avoirdupois.

TABLE XIX.

CONTAINING THE SUPERFICIES AND SOLID CONTENTS OF SPHERES, FROM ONE TO TWELVE, AND ADVANCING BY A TENTH.

Diam.	Superficies.	Solidity.	Diam.	Superficies.	Solidity.
1·0	3·1416	·5239	3·0	28·2744	14·1372
·1	3·8013	·6969	·1	30·1907	15·5985
·2	4·5239	·9047	·2	32·1699	17·1573
·3	5·3093	1·1503	·3	34·2120	18·8166
·4	6·1575	1·4367	·4	36·3168	20·5795
·5	7·0686	1·7671	·5	38·4846	22·4493
·6	8·0424	2·1466	·6	40·7151	24·4290
·7	9·0792	2·5724	·7	43·0085	26·5219
·8	10·1787	3·0536	·8	45·3647	28·7309
·9	11·3411	3·5913	·9	47·7837	31·0594
2·0	12·5664	4·1888	4·0	50·2656	33·5104
·1	13·8544	4·8490	·1	52·8102	36·0870
·2	15·2053	5·5752	·2	55·4178	38·7924
·3	16·6190	6·3706	·3	58·0881	41·6298
·4	18·0956	7·2382	·4	60·8213	44·6023
·5	19·6350	8·1812	·5	63·6174	47·7130
·6	21·2372	9·2027	·6	66·4782	50·9651
·7	22·9022	10·3060	·7	69·3979	54·3617
·8	24·6300	11·4940	·8	72·3824	57·9059
·9	26·4208	12·7700	·9	75·4298	61·6010

Diam.	Superficies.	Solidity.	Diam.	Superficies.	Solidity.
5·0	78·5400	65·4500	8·6	232·3527	333·0389
·1	81·7130	69·4560	·7	237·7877	344·7921
·2	84·9488	73·6223	·8	243·2855	356·8187
·3	88·2475	77·9519	·9	248·8461	369·1217
·4	91·6090	82·4481			
·5	95·0334	87·1139	9·0	254·4696	381·7044
·6	98·5205	91·9525	·1	260·1558	394·5697
·7	102·0705	96.9670	·2	265·9130	407·7210
·8	105·6834	102·1606	·3	271·7169	421·1613
·9	109·3590	107·5364	·4	277·5917	434·8937
			·5	283·5294	448·9215
6·0	113·0976	113·0976	·6	289·5298	463·2477
·1	116·8989	118·8472	·7	295·5931	477·7755
·2	120·7631	124·7885	·8	301·7192	492·8081
·3	124·6901	130·9246	·9	307·9082	508·0485
·4	128·6799	137·2585			
·5	132·7326	143·7936	10·0	314·1600	523.6000
·6	136·8480	150·5329	·1	320·4746	539·4656
·7	141·0264	157·4795	·2	326·8520	555·6485
·8	145·2675	164·6365	·3	333·2923	572·1518
·9	149·5715	172·0073	·4	339·7954	588·9784
			·5	346·3614	606·1324
7·0	153·9384	179·5948	·6	352·9901	623·6159
·1	158·3680	187·4021	·7	359·6817	641·4325
·2	162·8605	195·4326	·8	366·4362	659·5852
·3	167·4158	203·6893	·9	373·2539	678·0771
·4	172·0340	212·1752			
·5	176·7150	220·8937	11·0	380·1336	696·9116
·6	181·4588	229·8478	·1	387·0765	716·0915
·7	186·2654	239·0511	·2	394·0823	735·6200
·8	191·1349	248·4754	·3	401·1059	755·5008
·9	196·0672	258·1552	·4	408·2823	775·7364
			·5	415·4766	796·3301
8·0	201·0624	268·0832	·6	422·7336	817·2851
·1	206·1203	278·2625	·7	430·0536	838·6045
·2	211·2411	288.6962	·8	437·4363	860·2915
·3	216·4248	299·3876	·9	444·8819	882·3492
·4	221·6712	310·3398			
·5	226·9806	321·5558	12·0	452·3904	904·7808

This table will serve equally well for any and every unit of measurement. If the unit be inches, then the answer will be in inches : and so with feet, yards, metres, or miles. If the diameter of the earth be taken at 7,900 miles, (the mean diameter is really 7,914 miles), then the surface will be 196,067,200 square miles, and the contents 258,155,200,000 cubic miles, because the superficies are as the square, and the solid contents as the cube of the diameter. As 7,900 equals 1,000 times 7·9, then the figures opposite the latter must be multiplied by the square and by the cube of 1,000 respectively.

TABLE XX.

CONTAINING THE WEIGHT OF CAST-IRON PIPES, ONE FOOT LONG.

Diam. of bore in Inches.	THICKNESS IN INCHES.							
	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1-in.	$1\frac{1}{8}$	$1\frac{1}{4}$
	Pounds	Pounds	Pounds	Pounds	Pounds	Pounds	Pounds	Pounds
$1\frac{1}{2}$	6·9	9·9	......	......	......	......	......	......
2	8·8	12·3	16·1	20·3	......	......	......	......
$2\frac{1}{2}$	10·6	14·7	19·2	23·9	......	......	......	......
3	12·4	17·2	22·2	27·6	33·3	39·3	45·6	......
$3\frac{1}{2}$	14·2	19·6	25·3	31·3	37·6	44·2	51·1	......
4	16·8	22·1	28·4	35·0	41·9	49·1	56·6	64·4
$4\frac{1}{2}$	18·0	24·5	31·4	38·7	46·2	54·0	62·1	70·6
5	19·8	27·0	34·5	42·3	50·5	58·9	67·6	76·7
$5\frac{1}{2}$	21·6	29·5	37·6	46·0	54·8	63·8	73·2	82·8
6	23·5	31·9	40·7	49·7	59·1	68·7	78·7	88·8
$6\frac{1}{2}$	25·3	34·4	43·7	53·4	63·4	73·4	84·2	95·1
7	27·2	36·8	46·8	56·8	67·7	78·5	89·7	101·2
$7\frac{1}{2}$	29·0	39·1	49·9	60·7	72·0	83·5	95·3	107·4
8	30·8	41·7	52·9	64·4	76·2	88·4	100·8	113·5
$8\frac{1}{2}$	32·9	44·4	56·2	68·3	80·8	93·5	106·5	119·9
9	34·5	46·6	59·1	71·8	84·8	98·2	111·8	125·8
$9\frac{1}{2}$	36·3	49·1	62·1	75·5	89·1	103·1	117·4	131·9
10	38·2	51·5	65·2	79·2	93·4	108·0	122·8	138·1
$10\frac{1}{2}$	......	54·0	68·2	82·8	97·7	112·9	128·4	144·2
11	......	56·4	71·3	86·5	102.0	117·8	133·9	150·3
$11\frac{1}{2}$	......	58·9	74·3	90·1	106·3	122·7	139·4	156·4
12	......	61·3	77·4	93·6	110·6	127·6	145·0	162·6
13	......	......	82·7	101·2	118·2	137·4	154·1	173·5
14	......	......	89·3	108·2	126·5	146·2	165·3	185·2
15	......	......	95·2	115·7	135·3	156·2	176·2	198·1
16	.....	......	......	123·3	143·1	166·1	187·5	211·3
17	......	......	......	130·2	152·5	178·5	198·2	223·4
18	......	......	......	137·0	161·2	185·3	209·1	235·6
19	......	......	......	......	169·2	195·7	222·3	247·1
20	......	......	......	......	178·1	205·2	233·2	259·0
21	......	......	......	......	......	214·1	243·5	273·2
22	......	......	......	......	......	223·0	254·8	285·4
23	......	......	......	......	......	233·4	265·5	298.3
24	......	......	......	......	......	245·2	277·5	310·6

NOTE.—The area of a circle in inches, multiplied by the length in inches, and by ·263 = the weight in lbs. avoirdupois of cast-iron. By this rule any other dimensions may be calculated. To get the weight of a Pipe it is only necessary to ascertain the weight of the external and internal diameters, and subtract the latter from the former.

TABLE XXI.

SHOWING THE WEIGHT OF WATER IN PIPES OF VARIOUS DIAMETERS, ONE FOOT IN LENGTH.

Diameter in Inches.	Weight in Pounds.	Diameter in Inches.	Weight in Pounds.	Diameter in Inches.	Weight in Pounds.
3	3	12¼	51	23	180¼
3¼	3½	12½	53¼	23½	188¼
3½	4¼	12¾	55½	24	196¼
3¾	4¾	13	57½	24½	204½
4	5½	13¼	59¾	25	213
4¼	6¼	13½	62¼	25½	221½
4½	7	13¾	64½	26	230½
4¾	7¾	14	66¾	26½	239½
5	8½	14¼	69¼	27	248½
5¼	9¼	14½	71½	27½	257¾
5½	10¼	14¾	74¼	28	267¼
5¾	11¼	15	76¾	28½	276¾
6	12¼	15¼	79¼	29	286½
6¼	13¼	15½	82	29½	296½
6½	14¼	15¾	84½	30	306¾
6¾	15½	16	87¼	30½	317¼
7	16¾	16¼	90	31	327½
7¼	18	16½	92¼	31½	338¼
7½	19¼	16¾	95½	32	349
7¾	20½	17	98½	32½	360
8	21¾	17¼	101½	33	371¼
8¼	23¼	17½	104½	33½	382½
8½	24½	17¾	107½	34	394
8¾	26	18	110½	34½	405¾
9	27½	18¼	113½	35	417½
9¼	29¼	18½	116½	35½	429½
9½	30¾	18¾	119¾	36	441¾
9¾	32½	19	123	36½	454
10	34	19¼	126¼	37	466½
10¼	35½	19½	129½	37½	479¼
10½	37½	19¾	132	38	492¼
10¾	36¼	20	136¼	38½	505¼
11	41¼	20½	143¼	39	518½
11¼	43¼	21	150¼	39½	531¾
11½	45	21½	157½	40	545½
11¾	47	22	165		
12	49	22½	172½		

TABLE XXII.

[S]HOWING THE AMOUNT OF "LAP" REQUIRED FOR SLIDE VALVES, WHEN THE STEAM IS TO BE WORKED EXPANSIVELY.

Traverse of the Valve in inches.	The traverse of the piston where the steam is cut off.								
	$\frac{1}{4}$	$\frac{1}{3}$	$\frac{5}{12}$	$\frac{1}{2}$	$\frac{7}{12}$	$\frac{2}{3}$	$\frac{3}{4}$	$\frac{10}{12}$	$\frac{11}{12}$
	The required "lap."								
2	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{11}{16}$	$\frac{5}{8}$	$\frac{9}{16}$	$\frac{1}{2}$	$\frac{7}{16}$	$\frac{3}{8}$	$\frac{5}{16}$
$2\frac{1}{2}$	$1\frac{1}{16}$	1	$\frac{7}{8}$	$\frac{13}{16}$	$\frac{11}{16}$	$\frac{9}{16}$	$\frac{1}{2}$	$\frac{7}{16}$	$\frac{3}{8}$
3	$1\frac{1}{4}$	$1\frac{3}{16}$	$1\frac{1}{8}$	1	$\frac{15}{16}$	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{9}{16}$	$\frac{1}{2}$
$3\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{5}{16}$	$1\frac{3}{16}$	$1\frac{1}{8}$	$1\frac{1}{16}$	1	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{9}{16}$
4	$1\frac{3}{4}$	$1\frac{9}{16}$	$1\frac{7}{16}$	$1\frac{5}{16}$	$1\frac{1}{4}$	$1\frac{1}{16}$	1	$\frac{13}{16}$	$\frac{5}{8}$
$4\frac{1}{2}$	2	$1\frac{13}{16}$	$1\frac{9}{16}$	$1\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{1}{4}$	$1\frac{1}{8}$	$\frac{7}{8}$	$\frac{3}{4}$
5	$2\frac{1}{8}$	2	$1\frac{13}{16}$	$1\frac{9}{16}$	$1\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{1}{4}$	1	$\frac{13}{16}$
$5\frac{1}{2}$	$2\frac{5}{16}$	$2\frac{3}{16}$	2	$1\frac{13}{16}$	$1\frac{5}{8}$	$1\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{1}{8}$	$\frac{7}{8}$
6	$2\frac{1}{2}$	$2\frac{7}{16}$	$2\frac{3}{16}$	2	$1\frac{13}{16}$	$1\frac{5}{8}$	$1\frac{1}{2}$	$1\frac{3}{16}$	$\frac{15}{16}$
$6\frac{1}{2}$	$2\frac{3}{4}$	$2\frac{9}{16}$	$2\frac{7}{16}$	$2\frac{3}{32}$	2	$1\frac{13}{16}$	$1\frac{5}{8}$	$1\frac{1}{4}$	1
7	3	$2\frac{11}{16}$	$2\frac{9}{16}$	$2\frac{3}{8}$	$2\frac{3}{32}$	2	$1\frac{3}{4}$	$1\frac{7}{8}$	$1\frac{1}{8}$
$7\frac{1}{2}$	$3\frac{3}{16}$	3	$2\frac{11}{16}$	$2\frac{1}{2}$	$2\frac{3}{8}$	$2\frac{3}{32}$	$1\frac{7}{8}$	$1\frac{1}{2}$	$1\frac{3}{16}$
8	$3\frac{5}{16}$	$3\frac{3}{16}$	3	$2\frac{5}{8}$	$2\frac{1}{2}$	$2\frac{3}{8}$	2	$1\frac{5}{8}$	$1\frac{1}{4}$
$8\frac{1}{2}$	$3\frac{5}{8}$	$3\frac{5}{16}$	$3\frac{3}{16}$	$2\frac{13}{16}$	$2\frac{11}{16}$	$2\frac{1}{2}$	$2\frac{1}{8}$	$1\frac{3}{4}$	$1\frac{5}{16}$
9	$3\frac{13}{16}$	$3\frac{5}{8}$	$3\frac{5}{16}$	3	$2\frac{13}{16}$	$2\frac{11}{16}$	$2\frac{1}{4}$	$1\frac{7}{8}$	$1\frac{3}{8}$
$9\frac{1}{2}$	4	$3\frac{13}{16}$	$3\frac{5}{8}$	$3\frac{3}{16}$	3	$2\frac{13}{16}$	$2\frac{3}{8}$	2	$1\frac{7}{16}$
10	$4\frac{1}{4}$	4	$3\frac{13}{16}$	$3\frac{5}{16}$	$3\frac{3}{16}$	3	$2\frac{1}{2}$	$2\frac{1}{16}$	$1\frac{1}{2}$
$10\frac{1}{2}$	$4\frac{7}{16}$	$4\frac{1}{4}$	4	$3\frac{1}{2}$	$3\frac{5}{16}$	$3\frac{1}{8}$	$2\frac{5}{8}$	$2\frac{3}{16}$	$1\frac{9}{16}$
11	$4\frac{9}{16}$	$4\frac{7}{16}$	$4\frac{1}{4}$	$3\frac{5}{8}$	$3\frac{1}{2}$	$3\frac{3}{16}$	$2\frac{3}{4}$	$2\frac{1}{4}$	$1\frac{5}{8}$
$11\frac{1}{2}$	$4\frac{13}{16}$	$4\frac{9}{16}$	$4\frac{7}{16}$	$3\frac{7}{8}$	$3\frac{5}{8}$	$3\frac{3}{8}$	$2\frac{7}{8}$	$2\frac{3}{8}$	$1\frac{3}{4}$
12	5	$4\frac{13}{16}$	$4\frac{9}{16}$	$4\frac{1}{8}$	4	$3\frac{5}{8}$	3	$2\frac{1}{2}$	$1\frac{7}{8}$

The traverse of the valves being ascertained, and also the amount of cut-[o]ff desired, the above table shows the amount of "lap" required.

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